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An International Journal on Fundamentals of Friction, Lubrication, Wear, and their Control in Industry

WEAR

USURE VERSCHLEISS

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ON FUNDAMENTALS OF FRICTION, LUBRICATION, WEAR,
AND THEIR CONTROL IN INDUSTRY

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EDITORIAL

To promote discussions on "Control of Wear in Industry" is one of the aims of this Journal. However, in many industries wear is not yet well controlled and, moreover, accounts of experience on maintenance are not readily published. In this issue, reduction of wear in diesel engines is treated in an extensive report from a group of Danish shipowners. The operational analysis of wear phenomena was performed with the aid of an electronic computer. A remarkable detail of the operation is the discipline of all who participated in collecting the numerical information. This was done by controlling, over a number of years, the engines of 33 ships sailing the seven seas.

As a preliminary result, ways of obtaining more accurate information are indicated, for example separating "running-in" wear over a period of 3000 hours from the further wear history of the cylinder liners. It seems that corrosive wear, brought about by the formation of concentrated sulphuric acid, contributes considerably to the loss of material, as was suggested previously by FURSUND (see *Wear*, *1* (1957/58) 118). The plea of LEES for closer co-operation between interested parties might lead to an application of his statistical approach to the fleet of tankers operated by oil manufacturers. The contribution of oil constituents to corrosive cylinder wear is known in principle and needs only more detailed verification. The real unknowns are engineering and metallurgical factors.

Some practical means of reducing the surface temperature of cylinders — without necessitating major changes in construction — are reported in the Danish paper. It seems that this engineering approach leads to some reduction in the rate of corrosive wear. However, the combined evidence of FURSUND's and LEES's papers point to the fact that metallurgical properties of cylinder liners are never identical. Some method should be found for measuring the rate of wear due to instantaneous attack by sulphuric acid followed by the *mechanical* removal of this *chemically* weakened surface layer. If indeed wear rate is controlled by this mechano-chemical interaction, it is not surprising that the variability of wear resistance (Fig. 8, p. 282) is not correlated with conventional testing of materials.

The service trial reported extended over periods of $1–6 \cdot 10^4$ hours, time intervals long enough to make the future use of modern analytical tools attractive. As a tool, emission spectroscopy can sometimes compete with tracer techniques, as will be shown by Miss PIETSCHMANN in one of the next issues.

Everyone interested in diesel testing should consult the Annual Report of the Coordinating Research Council (see bibliographies of this issue). The list of names of industrial and U.S. government agencies involved in diesel evaluation problems is rather impressive. The bulk of such committee work seems to aim at *performance* testing. It may easily be that the *manufacture* of cylinder liners with a reproducible corrosive wear rating — a metallurgical problem — will need a different and still more sophisticated, analytical approach.

Delft, March 1959 G. SALOMON

LUBRICANT BEHAVIOUR IN CONCENTRATED CONTACT SYSTEMS — THE CASTOR OIL–STEEL SYSTEM

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SUMMARY

Experiments are described in which frictional forces are measured between hardened steel surfaces lubricated with castor oil. In one series of experiments, balls of different sizes slide against a moving plane at a maximum Hertz stress of 389,000 p.s.i. (273.5 kg/mm^2) over a range of speeds up to 140 cm/sec. In another experiment the frictional force is measured between rollers which roll with a small component of sliding motion, the elastic stresses being similar to those used in the first experiment.

In an interpretation of these measurements, it is concluded that in a certain range of speed most of the frictional force is transmitted through a film of lubricant acting as a plastic solid whose shear strength decreases with increasing temperature, and that in the sliding experiment the coefficient of friction gives essentially a measure of the temperature rise in the lubricant.

It is further suggested that in concentrated contact systems of this type where the frictional force is determined by the high-stress plastic yield of the lubricant, the film thickness is determined by viscous flow properties.

ZUSAMMENFASSUNG

VERHALTEN DES SCHMIERMITTELS IN SYSTEMEN MIT KONZENTRIERTEM KONTAKT. DAS SYSTEM RIZINUSÖL–STAHL

Die Messung von Reibungskräften zwischen gehärteten Stahlplatten, die mit Rizinusöl geschmiert wurden, wird beschrieben. In einer Gruppe von Versuchen gleiten Kugeln verschiedener Grösse auf einer sich bewegenden Fläche, wobei maximale Hertz-Spannungen von 273.5 kg/mm^2 auftreten und zwar mit zunehmenden Geschwindigkeiten bis zu 140 cm/sec. In einer anderen Versuchsanordnung wurden die Reibungskräfte zwischen Rollen gemessen, wobei die Rollbewegung eine kleine Komponente des Gleitens besass, die elastischen Spannungen waren hierbei ähnlich wie die beim ersten Experiment.

Die Ergebnisse werden wie folgt gedeutet: Innerhalb gewisser Grenzen der Geschwindigkeit werden die Reibungskräfte durch den Schmierfilm übertragen; dieser verhält sich hierbei wie ein plastischer Festkörper, dessen Schubstärke mit zunehmender Temperatur abnimmt. Der Reibungskoeffizient im Gleitversuch ist hauptsächlich ein Mass für die Zunahme der Temperatur im Schmiermittel.

Weiterhin wird angenommen, dass in solchen konzentrierten Kontaktsystemen, in denen die Reibungskräfte durch das plastische Fliessen des Schmiermittels unter hoher Spannung bestimmt werden, die Dicke dieses Films von den viskosen Eigenschaften abhängt.

LIST OF SYMBOLS

H Rate of flow of heat per unit area

k Thermal conductivity

$N_H = \frac{pR}{\eta U}$ The Hersey number

$N_N = \frac{HR}{k\theta}$	The Nusselt number
N'_N	A dimensionless parameter
p	Pressure or normal contact stress
P_y	Tangential force
P_z	Normal load
R	Radius of a sphere; a characteristic length
U	Surface speed; a characteristic velocity
α	Angle between the axes of skewed rollers
η	Viscosity
μ	Coefficient of friction
θ	Temperature
τ_y	Yield stress of a plastic solid

INTRODUCTION

The experiments described here are concerned with certain aspects of the way in which lubricants behave in components such as ball bearings and gears, where concentrated loads act on the regions of contact.

The selection of lubricants for these applications is sometimes difficult, especially for high-temperature service. There seems to be a lack of knowledge about the mechanical properties and rheological behaviour of lubricants under the high stresses found in contact zones. There is also uncertainty in relating the performance of a lubricant in a machine in service to its behaviour in simple laboratory rolling or sliding devices. The essential requirement here is to determine the conditions for similitude in the behaviour of a lubricant in a series of concentrated contact systems.

These problems are approached in an elementary way by studying firstly a series of systems in which balls of different sizes slide on a lubricated plane. Secondly, the behaviour of these systems is compared with that of a rolling system under similar contact stress. In the rolling system a force is measured which is transmitted through a lubricant film between rollers whose axes are skewed, causing the rolling motion to be accompanied by a component of sliding motion. In both experiments, ball bearing steel components were used with castor oil as the lubricant.

SLIDING FRICTION EXPERIMENTS

The machine used for sliding friction measurements is shown diagrammatically in Fig. 1. A steel specimen (SAE 52100 steel, hardened to Rockwell C 61 to 63) is rotated in a horizontal plane by an adjustable speed motor. The slider is a steel bearing ball of 1/4, 1/2 or 1 in. diameter, mounted so as to touch the plane on a radius of 1/2 in., giving sliding speeds of up to 140 cm/sec. A deadweight load can be applied to the specimen. The loads used and the resulting calculated stress and contact zone dimensions are shown in Table I.

The friction force is measured with a strain gauge device. To increase the accuracy

of measurement, the friction force can be partly balanced by weights acting over pulleys.

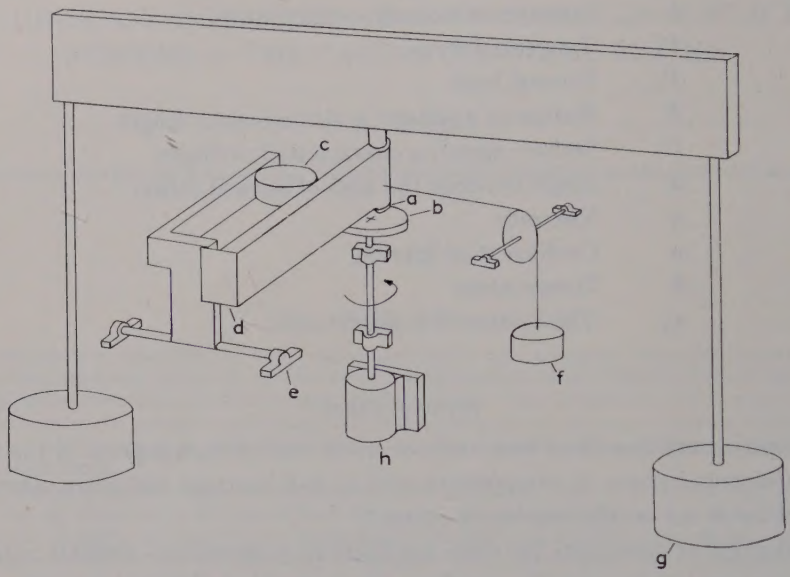


Fig. 1. Diagram of sliding friction machine.

a Spherical slider
b Rotating plane
c Strain gauge dynamometer
d Flexure strip
e Pivot
f Weight to partly balance frictional force
g Normal load
h Motor

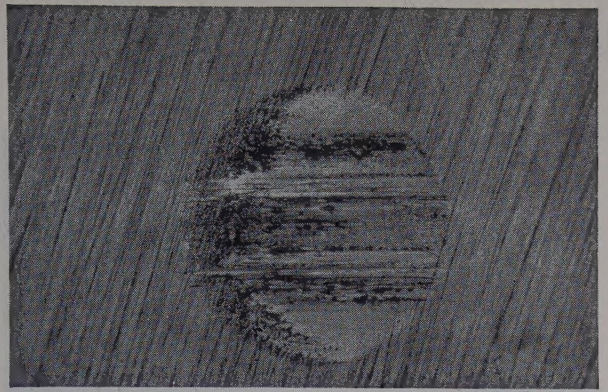
TABLE I

CALCULATED HERTZ MAXIMUM CONTACT STRESS AND CONTACT ZONE DIMENSIONS

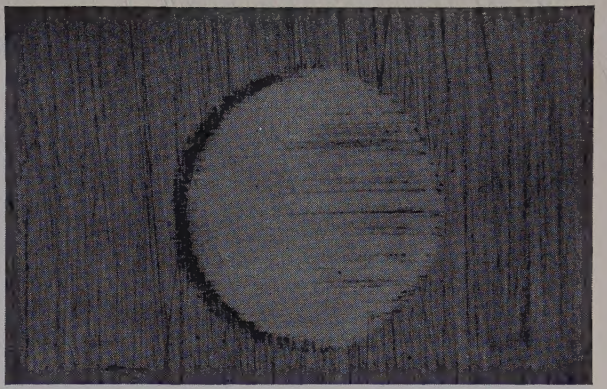
System		*Normal load*	*Max. contact stress*		*Major dia.*		*Minor dia.*	
		P_z(kg)	p.s.i.	kg/mm²	inch	mm	inch	mm
1 in. dia. ball	plane	128	389.000	273.5	0.0372	0.945	—	—
½ in. dia. ball	plane	32	389,000	273.5	0.0186	0.472	—	—
¼ in. dia. ball	plane	8	389,000	273.5	0.0093	0.236	—	—
2 in. dia. sphere	2 in. dia. cylinder	250	389,000	273.5	0.0654	1.661	0.0412	1.046
		62.5	246,000	173	0.0412	1.046	0.0260	0.660

For most of the experiments, the balls and planes were finished with 4/0 emery paper to a surface finish of 3.3 μin. C.L.A. (center line average). To render wear marks readily visible, the direction of abrasion of the ball was approximately at right angles to the direction of sliding. In some experiments the parts were finished with 1 to 3 μ diamond compound to 0.8 μin. C.L.A.

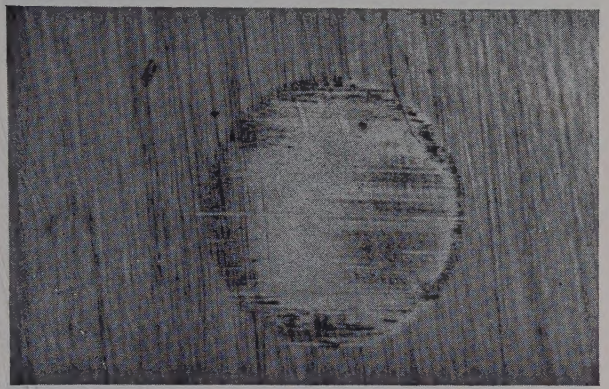
The castor oil, whose viscosity was 4.73 poise at 87°F (30.6°C) was applied to the ball at a rate of about 3 cm² per minute.



(a) Speed 0.08 cm/sec, distance 10.5 m.



(b) Speed 4.15 cm/sec, distance 41 m.



(c) Speed 17 cm/sec, distance 41 m.

(d) Speed 17 cm/sec, distance 3740 m.

Fig. 2. Contact zones of 1/2 in. steel balls, load 32 kg, lubricant castor oil.

(Magnification 70× ; 128× on original prints)

(a) Speed 17 cm/sec, distance 40 cm.

(b) Speed 69 cm/sec, distance 41 m.

(c) Speed 139 cm/sec, immediate seizure, diam. of scar 3 mm.

(d) Speed 140 cm/sec, distance 41 m.

Fig. 3. Contact zones of 1/2 in. steel balls, load 32 kg, lubricant castor oil.

(Magnification 70× ; 128× on original prints)

The experimental procedure was to lower the ball on to the rotating plane, to allow it to slide a fixed distance (in most cases, approximately 20, 40 and 80 metres respectively for the 1/4, 1/2 and 1 in. balls) and then to remove it. The system was never started or stopped under load.

The friction force was measured during this period, and after the experiment the contact zone of the ball was photographed under the microscope.

SLIDING FRICTION RESULTS

The appearance of the contact zones of some 1/2 in. balls run in this way at 389,000 p.s.i. maximum Hertz stress is shown in Figs. 2 and 3, with the leading edge of the zone towards the left in each case. At low speeds, below about 1 cm/sec, "stick-slip" occurs and the plane is scratched. Metal is picked up by the ball, but contact is confined to the Hertz contact circle. At speeds of from 1 to 4 cm/sec the sliding is smooth and less damage occurs to the plane. The contact zone is polished, with a dark crescent, presumably of fine wear products, at the leading edge. At higher speeds the front and centre of the zones suffer less wear until at speeds of from 40 to 140 cm/sec contact is confined to the sides and rear of the zone.

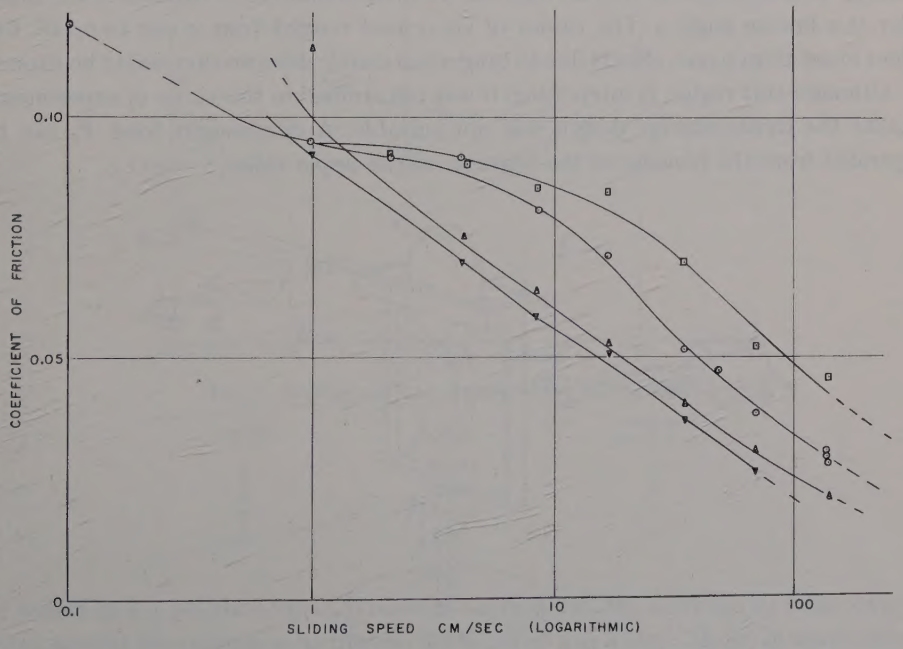


Fig. 4. Sliding friction of ball on plane. Lubricant castor oil.

Symbol	*Ball dia. (in.)*	*Load (kg)*	*Surface finish (μin. C.L.A.)*
⊙	1/2	32	3·3
△	1	128	3·3
▽	1	128	0.8
⊡	1/4	8	3·3

At 140 cm/sec, the behaviour was less consistent. In some experiments, very little wear occurred. In others, such as Fig. 3d incipient seizures occurred while in others there was a complete and destructive seizure (Fig. 3c). No friction measurements were made when complete seizure took place.

The effect of varying the distance of travel is shown for 1/2 in. balls sliding at 17 cm/sec. At 40 cm travel (Fig. 3a) contact seems to begin at the outer edges of the zone. After 3740 m (2d) no more wear has occurred than after 41 m (2c) although more solid material has been collected by the contact.

The appearance of the wear scars on the 1/4 in. and 1 in. balls was similar to that of the 1/2 in. balls.

The coefficients of friction measured in these experiments are shown in Fig. 4 plotted against the logarithm of the speed. The running-in effect was not marked and the sliding distances used sufficed for the friction force to reach a steady value.

ROLLING CONTACT EXPERIMENT

The device used for rolling contact experiments is shown diagrammatically in Fig. 5. The lower cylindrical roller (of 2 in. diameter) is driven by a motor at a surface speed U. The upper roller has a spherical surface of 1 in. radius and is free to rotate in ball bearings. It is arranged so that its axis can be skewed relative to the axis of the lower roller at a known angle α. The values of $\sin \alpha$ used ranged from 0.002 to 0.128. For values lower than 0.002, effects due to tangential elastic deformation would be expected. Although this region is interesting, it was not studied in this series of experiments because the dynamometer design was not suitable. A deadweight load P_z can be suspended from the housing of the bearings of the upper roller.

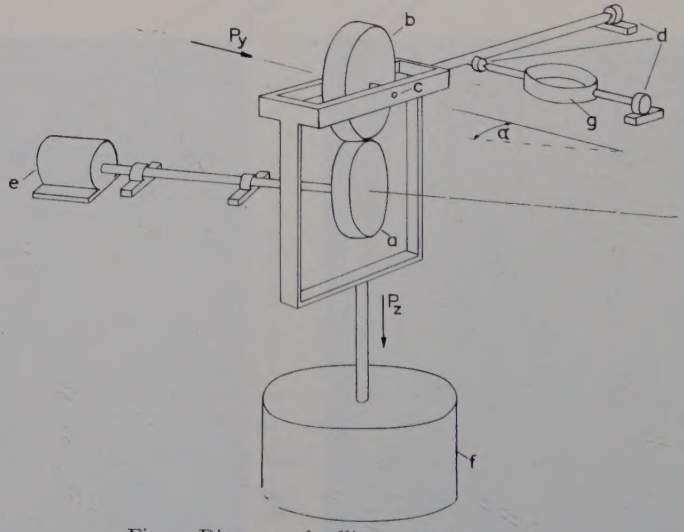


Fig. 5. Diagram of rolling contact machine.

a Cylindrical roller
b Spherical roller
c Bearings
d Pivots
e Motor
f Normal load
g Strain gauge dynamometer

The skewing of the rollers causes an axial force P_y to act on the upper roller. This force is measured with a strain gauge dynamometer. The force P_y is interpreted as the component of force required to shear the contact zone with a sliding velocity $U \sin \alpha$ in the direction of the y axis, when this sliding motion is superposed on a rolling velocity U.

Results are presented in Figs. 6 and 7 for two loads, 250 kg and 62.5 kg, giving the stresses and contact zone diameters shown in Table I. In most of the experiments

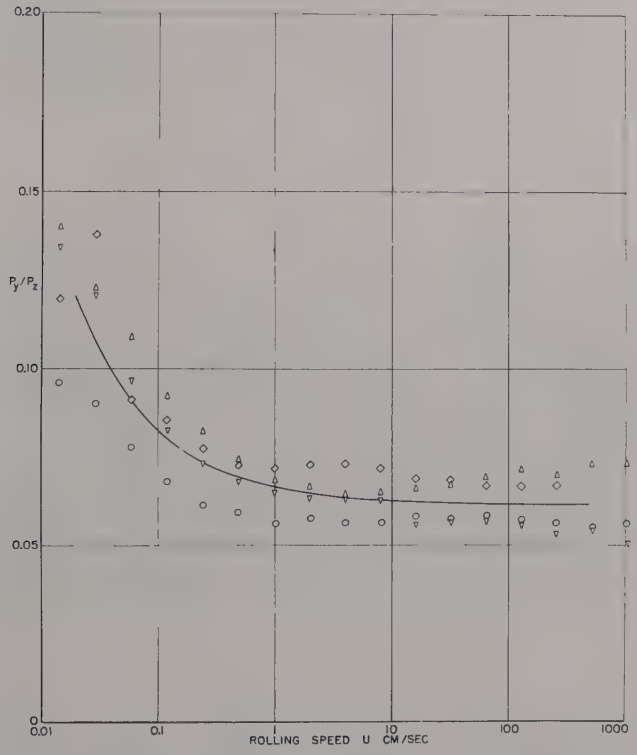


Fig. 6. Rolling contact experiment – 250 kg load.

Symbol	*Sin α*	*Roughness (μin. C.L.A.)*
○	0.002	6
△	0.008	6
▽	0.032	6
◇	0.28	6

the rollers had a polished finish. Traces of surface damage occurred in some experiments, so that the roughness at the end of the series was 6 μin. C.L.A. In some experiments at the lighter load the lower roller was replaced by ones treated with grade 1 emery paper and with grade 1 1/2 emery cloth, giving C.L.A. values of 33 and 67 μ in. respectively. Profile traces of these surfaces are shown in Fig. 8.

At speeds above 200 cm/sec the value of P_y fluctuated in an apparently erratic manner. Only the region below this speed will be considered in the interpretation.

INTERPRETATION

Both the sliding and the rolling experiments indicate the existence, in a certain range of speeds, of films of lubricant which are thicker than the height of the surface roughness and which extend over most of the area of contact. Although at the lowest speeds this is not true, and the demarcation is not defined, it will be assumed here that the data selected for analysis refer to a lubricant film which behaves as a mechanical

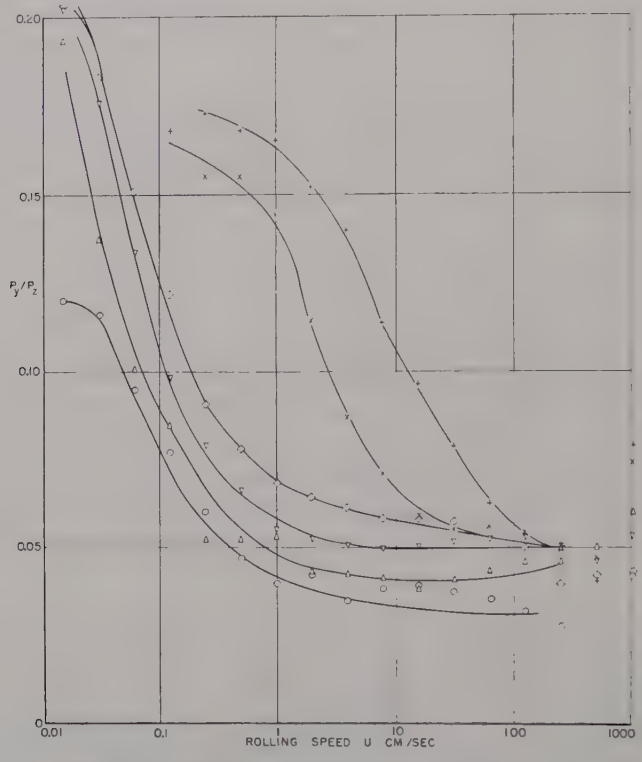
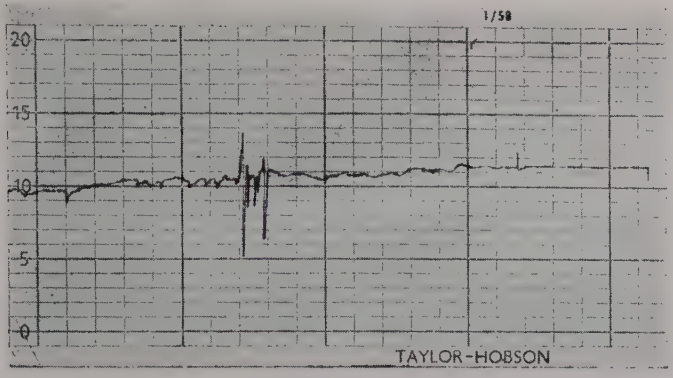


Fig. 7. Rolling contact experiment – 62.5 kg load.

Symbol	*Sin α*	*Roughness μin. C.L.A.*
o	0.002	6
Δ	0.008	6
∇	0.032	6
◇	0.128	6
×	0.032	33
+	0.032	67

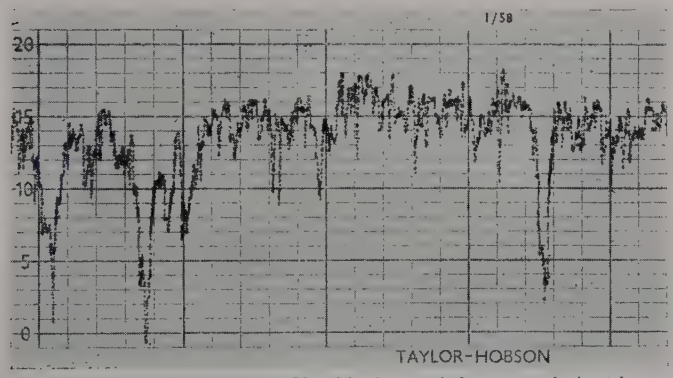
continuum rather than as a molecular layer. The principal property considered is the frictional force which opposes a sliding motion. The frictional force is interpreted as a shearing force acting on the lubricant film, and as a measure of the shear stress in the film. In the sliding experiments the frictional force is given as the coefficient of friction, and in the rolling experiments as the ratio P_y/P_z of axial force to normal force.

The evidence about film thickness in the rolling experiments comes from the fact



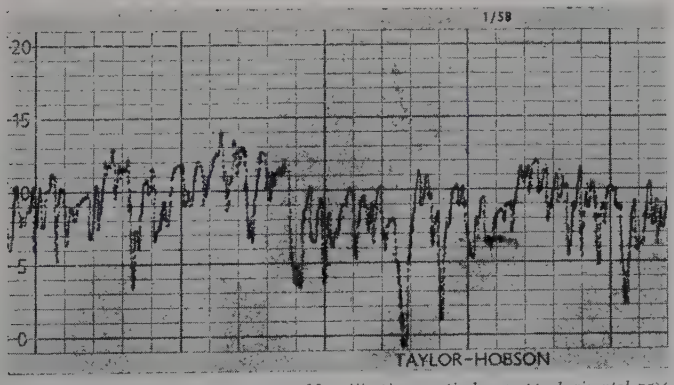
Magnification: vertical 35,000×, horizontal 75×

Polished surface (after experiment)
C.L.A. = 6 μin.



Magnification: vertical 15,000×, horizontal 75×

Surface treated with # 1 emery paper
C.L.A. = 33 μin.



Magnification: vertical 7,500×, horizontal 75×

Surface treated with # $1^1/_2$ emery cloth
C.L.A. = 67 μin.

Fig. 8. Rolling contact experiment – surface profiles.

that rough surfaces tend at the higher speeds to give frictional behaviour similar to that of smooth surfaces. In the sliding experiments the evidence is photographic: the original surface finish of 3.3 μin. C.L.A. is undisturbed over the greater part of the contact zone at speeds above 17 cm/sec.

The crescent-shaped wear scar found at speeds in the region of 67 cm/sec indicates an "elastohydrodynamic" effect in which the bodies are elastically deformed by fluid pressure. No analysis of the geometric evidence provided by these photographs is attempted at present.

The present interpretation is based upon the following three observations on the trend of the measured frictional forces.

(a) In the rolling experiment (Fig. 6) at 389,000 p.s.i. maximum Hertz stress, the shear stress in the film (as given by P_y/P_z) is independent of the rolling speed U and of the sliding velocity $U \sin \alpha$ over a wide range, despite the increase in film thickness with speed suggested by the results with rough surfaces in Fig. 7.

(b) In the sliding experiments at 389,000 p.s.i. the shear stress decreases with speed to a value lower than the shear stress in the corresponding rolling experiment at 389,000 p.s.i.

(c) In the series of sliding experiments with balls of different sizes at the same contact stress, the shear stress is the same in two systems when the larger system is sliding at a lower speed than the smaller one.

The first observation suggests that the process in the contact zone for a predominantly rolling motion may involve the deformation of a plastic solid whose shear strength is independent of the rate of shear. The second observation indicates that if in the purely sliding process the lubricant behaves as a plastic solid, it is as a plastic solid whose shear strength decreases with the sliding speed.

The implied contradiction between these two observations can be removed if it is assumed that the shear strength of the lubricant as a plastic solid is dependent not upon the rate of shear but upon the temperature rise resulting from an increase in sliding speed. This hypothesis can be investigated by examining the conditions for similitude in several simplified models of the sliding process and checking them with observation (c) above.

In this technique, one studies experimentally the properties of a series of systems which are geometrically similar in order to determine the functional relation which governs the variation of certain parameters while certain other characteristics remain constant. This functional relation is compared with those given by dimensional analysis of the theoretical models. In the present case we are concerned with the relation between the speed and radius of a series of lubricated sliding spheres when the coefficient of friction is constant.

The first model to be considered in this way is the conventional hydrodynamic lubrication process in which the lubricant is characterized by a uniform viscosity so that the shear stress at any point in the system is proportional to the rate of shear at that point. Systems conforming to this model are characterized by the Hersey number N_H:

$$N_H = \frac{pR}{\eta U} \tag{1}$$

If the Hersey number is the same in a geometrically similar series of such systems, then all other relevant dimensionless ratios, and in particular the coefficient of friction, will be the same in each member of the series. If the coefficient of friction is plotted against U/R the points for systems of all sizes should fall on the same line.

In the present experiments, an opposite type of relation is found, since with μ constant R decreases as U increases. The discrepancy is shown by the plot of μ against U/R in Fig. 9. It may be deduced from this that in the sliding process the shear stress is not even approximately proportional to the rate of shear as the size of the system is varied.

The second model of a sliding process to be considered is an entirely plastic one, in which the shear occurs, at a rate independent of the stress, when the shear stress

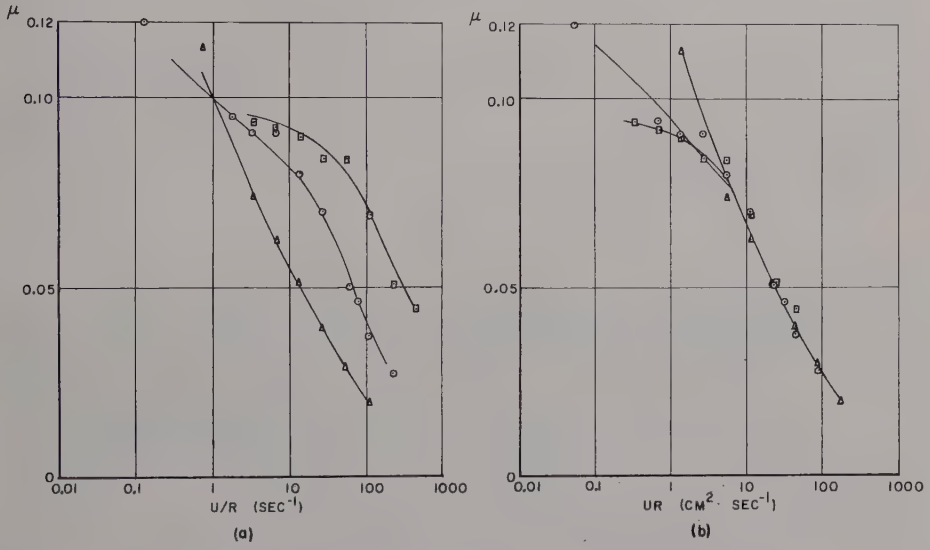


Fig. 9. Data on Fig. 4 re-plotted showing (a) correlation based on Hersey number (b) correlation based on Nusselt number. Symbols as in Fig. 4.

at any point reaches a certain value — the yield stress. It will be assumed secondly that the yield stress at any point depends on the temperature and pressure at the point, and therefore on the temperature rise caused by the frictional heat produced. Thirdly, it will be assumed that the magnitude of the temperature rise is determined by the rate at which heat can be conducted through the lubricant film. A series of geometrically similar systems, containing the same lubricant and with the same boundary conditions (distribution and temperatures of "sinks" of heat) can, if they conform to this model, be characterized by the Nusselt number:

$$N_N = \frac{HR}{k\theta} \tag{2}$$

Since the only source of heat in the system is the sliding process, the flow of heat H per unit area at a particular point may be expressed as the product of yield stress at the point and sliding velocity:

$$H = \tau_y U \tag{3}$$

It was assumed that the yield stress at a point depends only upon the temperature and pressure at the point. The temperature can therefore be expressed as a function of yield stress and pressure:

$$\theta = \theta(\tau_y, p). \tag{4}$$

The Nusselt number in a system conforming to the model may be re-written:

$$N_N = \frac{\tau_y UR}{k\theta(\tau_y, p)} \tag{5}$$

The distribution of yield stress values over the contact zone is a thermal property which will be the same for systems of the same N_N. To make further use of this parameter, it is necessary to restrict consideration to a series of systems in which the coefficient of friction is the same in each case. Yield stress at a point may then be replaced by the product of pressure and coefficient of friction for the process in an expression for the characteristic parameter of the system:

$$N'_N = \frac{\mu p UR}{k\theta(\mu, p)} \tag{6}$$

This parameter indicates that in a series of systems such as those studied here, U should vary inversely with R while the coefficient of friction μ and the characteristic stress p are held constant; when μ is plotted against UR for a series of systems of different size loaded to the same stress, the points should fall on a single line. Such a plot is shown in Fig. 9; a measure of agreement exists. Such agreement cannot "prove" the existence of plastic flow. It indicates that such a mechanism is not impossible and justifies further examination of plastic flow mechanisms.

A third model which can be considered in similar fashion is one in which temperature-dependent plastic flow occurs, but in which the heat of the shearing process is carried away by the thermal capacity of the flow of fluid through the contact zone rather than by thermal conductivity. It can be shown by similar reasoning that in a series of such geometrically similar systems with the same coefficient of friction, the radius of the ball will be independent of the speed.

FILM THICKNESS

Although the second model considered above is capable of explaining the observed trend in friction forces, it does not seem to lead readily to a determinate film thickness. This is because the temperature-dependence of the yield stress will cause the shearing to be restricted to a single shear plane—a situation with no element of convergent flow (Fig. 10a). Although other mechanisms may be involved, it seems preferable, because of the importance generally accorded to viscosity and viscosity–pressure

relations in lubrication, to consider ways in which convergent viscous flow may determine the film thickness. Two mechanisms are considered: they may be coexistent.

Firstly, the lubricant is a viscous liquid as it enters the leading edge of the zone and remains so until it reaches the high-pressure region where under the combined normal and shear stress it acts as a plastic solid. It is usually assumed that its behaviour in the intermediate range, where the increase of viscosity with pressure is important, is the significant factor in determining the thickness of the film as it passes into the high-pressure region.

Secondly, viscous flow may be occurring in the high-pressure region in addition to the plastic flow postulated in the model, but to such an extent as to make only a minor contribution to the energy loss. This is shown diagrammatically in Fig. 10b. The rate of shear in the viscous-flow region would be determined not as in most lubrication problems, by the sliding speed, but by the shear stress transmitted across

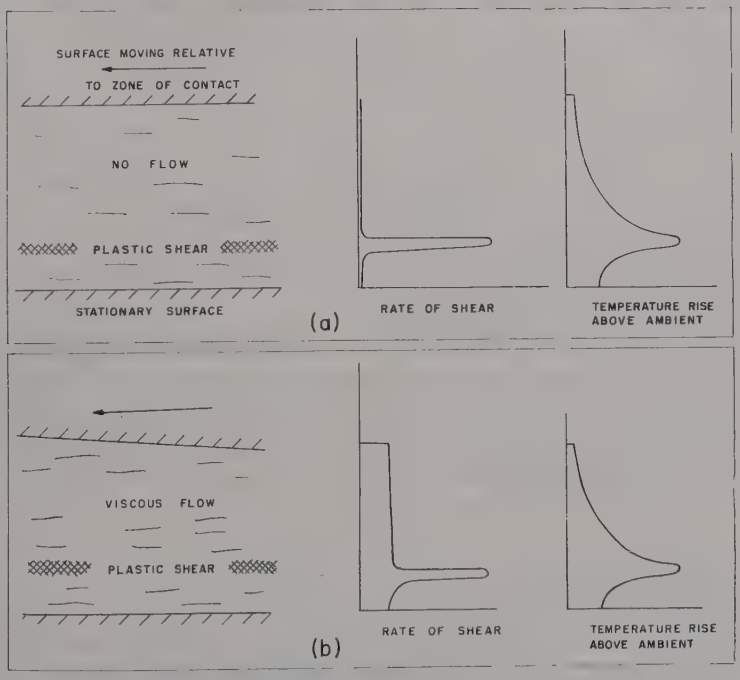


Fig. 10. Schematic diagrams of hypothetical flow processes for sliding in concentrated contact: (a) plastic shear; (b) combined viscous flow and plastic shear.

the plastic shear plane. In the present experiments with castor oil, this shear stress decreases with increasing sliding speed. This will in turn cause the rate of shear in the viscous-flow region, and hence its contribution to the film thickness, to decrease with increasing sliding speed. It is suggested that an effect of this sort may be responsible for the tendency to seizure which was found at high speeds with the castor oil–steel system.

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TIME EFFECTS IN THE STATIC FRICTION OF LUBRICATED RUBBER

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SUMMARY

The considerable increases in static friction that occur when lubricated rubber remains at rest under load are shown to be due to squeezing out of the lubricant film from between the surfaces.

Unlike the case of rigid materials, rubber continues to squeeze out the lubricant long after the surfaces have come into contact, and under certain conditions the process continues until the friction has increased to a value corresponding to unlubricated sliding.

The time-scale of the phenomenon appears to be directly related to the bulk viscosity of the lubricant and to the elastic modulus of the rubber. With rough surfaces the final friction value is greatly reduced.

ZUSAMMENFASSUNG

ZEITABHÄNGIGKEIT DER STATISCHEN REIBUNG VON GESCHMIERTEM GUMMI

Die statische Reibung von geschmiertem Gummi, der unter Belastung steht, nimmt mit der Zeit wesentlich zu. Es wird nachgewiesen, dass diese Erscheinung auf dem Herausdrücken des Schmierfilms zwischen den Oberflächen beruht.

Im Gegensatz zu Festkörpern drückt Gummi das Schmiermittel noch lange Zeit nachdem die Oberflächen Kontakt gemacht haben heraus; unter bestimmten Bedingungen setzt sich dieser Prozess fort bis die Reibung einen Wert, der dem des ungeschmierten Gleitens entspricht, erreicht hat.

Die Zeitabhängigkeit der Erscheinungen steht in unmittelbarer Beziehung zur Viskosität des Schmiermittels und zu dem Elastizitätsmodul des Gummis. Der Endwert der Reibung wird durch rauhe Oberflächen stark erniedrigt.

INTRODUCTION

Under certain conditions the static friction of lubricated metal surfaces may increase with time. For instance force fits become tighter if left for a period. This effect is usually attributed to squeezing out of the lubricant initially trapped between the surfaces, the static friction increasing until the whole load is supported by solid contact.

Such changes in static friction are particularly pronounced when one of the surfaces is highly elastic, as with rubber bearings and seals, where the effect is known as "stiction". DENNY[1] and MCCUISTON *et al.*[2] have shown that after sliding ceases the static friction of rubber seals may double within a few minutes and continue increasing for a period of several days. CLAYTON[3] suggested that the ability of rubber to conform to the contour of the rigid surface would tend to allow escape of lubricant even when the film thickness was extremely small, and would explain the fact that the process may extend over a very long period. Sliding under film conditions is common even with

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fluid seals so that there is usually no difficulty in explaining the initial separation of the surfaces by lubricant.

The present series of experiments was carried out during 1952–3 to determine whether the increase in static friction of rubber materials was consistent with the squeezing out of lubricant, and to determine the factors affecting the process.

EXPERIMENTAL PROCEDURE

The static friction of various rubber compounds in conjunction with different track surfaces and lubricants was measured after various periods of time during which the surfaces remained at rest under load.

In most of the experiments the rubber samples consisted of discs 0.48 cm in diameter cut from sheet 0.32 cm thick that had been moulded against polished steel. They were fitted into recesses in the loading plate, so that about 0.01 cm protruded from the under face; samples larger than 1 cm diameter were merely bonded to the loading plate. In each test three similar samples were used, located at the apexes of a triangle of 8 cm side.

For the unlubricated tests, the rubber samples were placed on the track and loaded within an hour of cleaning with acetone, to avoid blooming of the surface, since ROTH *et al.*[4] found that the presence of bloom affected the coefficient of friction. In the lubricated tests the track surfaces were flooded with lubricant before the samples were placed in position. After loading, the whole assembly was left for periods ranging from 15 sec to 3 months. To expedite the investigation several identical assemblies were in use at the same time.

The towing force was provided by a loaded three-wire linkage, as described in a previous paper (DENNY[5]), the horizontal wire being attached to the loading plate in the plane of the sliding surface. With this apparatus the applied force could be measured with an accuracy of $\pm$ 1% between 0.02 and 30 kg. Detection of the start of sliding was made difficult by elastic distortions of the samples, but with the aid of a low-power microscope directed at the interface, sliding could be observed before the samples had travelled 10^{-3} cm.

Details of the lubricants, rubbers and tracks used are given in Tables I-III.

LIST OF SYMBOLS

T	Time after applying load
η	Viscosity of lubricant
E	Compression modulus of rubber (applied stress/linear strain) for strains below 10%
W	Load
μ_0	Coefficient of static friction at zero time
μ_T	Coefficient of static friction at time T
$\hat{\mu}_T$	Coefficient of static friction after infinite time
μ_d	Coefficient of static friction under unlubricated conditions
R	Radius of specimen

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TABLE I

DETAILS OF TRACK SURFACES

Track	*Method of preparation*	*C.L.A. Roughness (μ)* (1)	(2)
Steel A	Hand lapped after grinding	0.06	0.06
B	Unindirectional abrasion with No. 1 emery cloth	0.20	0.38
C	Random abrasion with No. 1 emery cloth	0.15	0.18
D	Random abrasion with No. 2 emery cloth	0.32	0.38
Perspex A	Moulded against plate glass (as supplied)	0.01	0.01
B	Unidirectional abrasion with No. 1 emery cloth	0.38	1.37
Glass A	Plate glass as supplied	0.00	0.00
B	Ground glass as supplied	0.98	0.98
C.L.A.	(1) along finishing marks (2) across finishing marks		

TABLE II

PROPERTIES OF LUBRICANTS USED

Lubricant		*Approx. viscosity at 15°C (centipoise)*
A	Water	1.0
B	Light mineral oil (DTD 585)	25
C	DTD 585 + 2% colloidal graphite	25
D	Mineral oil	100
E	Mineral oil	410
F	Mineral oil	1250

TABLE III

PROPERTIES OF RUBBER SAMPLES USED

Sample	*Material*	*Hardness (BS degrees)*	*Compression Modulus E ($kgcm^{-2}$)*
A	Hycar synthetic rubber	60	49
B	Hycar synthetic rubber	90	310
C	Neoprene synthetic rubber	40	5.5
D	Hycar. Surface abraded on emery cloth	60	49

EXPERIMENTAL RESULTS

With unlubricated surfaces the static friction appeared to be independent of the time under load and no changes were recorded for periods up to several months. Effects similar to those described by SCHALLAMACH[6], which have been attributed to plastic deformation of the sample, were not encountered. Under lubricated conditions the static friction invariably increased considerably. The rate of increase, initially high, later diminished until ultimately, after a period that varied in the tests between a few hours and a few weeks, the friction reached a constant value.

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In the period immediately following application of the load, it was clear that the surfaces were separated by a continuous lubricant film, for no static friction was measurable. The time after loading at which static friction first appeared varied with load and viscosity in accordance with hydrodynamic theory, and corresponded to a value of $TW/\eta R^4$ of the order of $5 \cdot 10^{-7}$ cm^{-2}. This figure, incidentally, represents a film thickness of about 10^{-4} cm. The subsequent rise in static friction was found to undergo a considerable reduction in rate after a period some 100 times as long as the "floating" period, as shown in the typical results in Fig. 1. The initial rapid rise in friction corresponds to the case of loaded rigid surfaces, which has been studied in detail by NEEDS[7], MARDLES AND BIELAK[8] and others. In bearing and seal applications, where loads are relatively high and bearing areas small, the duration of this effect is likely to be so short as to be of no practical significance. Accordingly for the purposes of correlating the results of the present investigation, the duration of the initial period has been ignored and the friction value corresponding to the change in regime taken as the static friction at zero time.

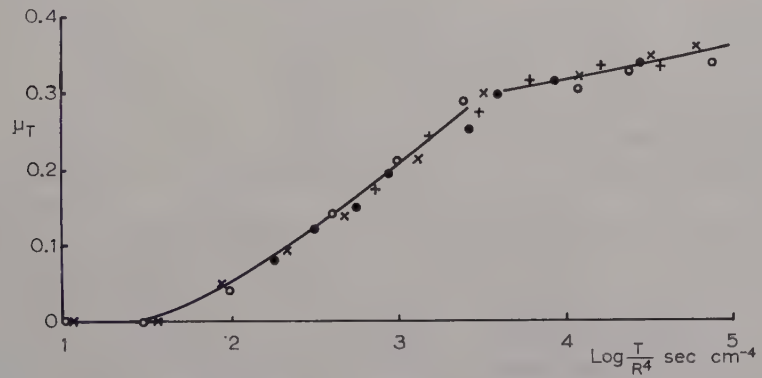

Fig. 1. Influence of time on static friction of lubricated rubber. $W = 1.13$ kg; Lubricant D (Mineral Oil); Rubber A (Hycar); Steel surface A.

× $R = 1.27$ cm + $R = 0.42$ cm
○ $R = 0.63$ cm ● $R = 0.32$ cm

The second, less rapid, stage of friction rise is peculiar to highly elastic materials. With smooth tracks the fraction was found ultimately to approach a value corresponding to unlubricated conditions, and inspection of the surfaces after a test showed that they could not be distinguished visually from freshly cleaned surfaces. In view of this finding, all the results presented have been expressed in terms of the equivalent unlubricated friction value.

A few experiments were carried out to discover what happened to the static friction when sliding was continued after breakaway. For sliding speeds up to 10^{-1} cm sec^{-1} the static friction gradually returned to its initial value provided lubricant was available on the track. The velocity coefficient tended to be of appreciable magnitude, and

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friction values much higher than the static figure were recorded; however, this aspect was not studied in detail.

It was found that if after standing under load the samples were lifted from the track and then returned to their former position, the appropriate high value of static friction was recorded. On the other hand, addition of lubricant to the track before replacing the samples resulted in friction values corresponding to the initial condition. No evidence of chemical attack, apart from occasional discolouration of the steel surfaces, was apparent at any time.

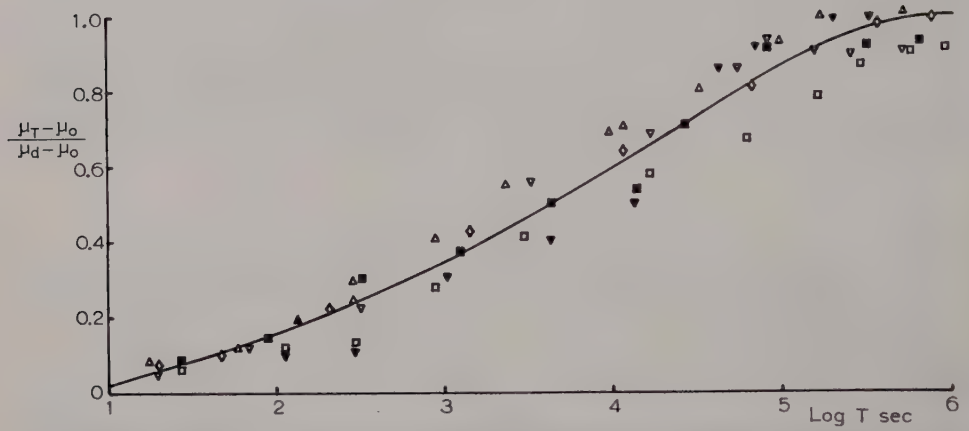

Fig. 2. Effect of load and track material, for smooth track surfaces. Lubricant A (water); Sample B (Hycar); $R = 0.24$ cm.

	Track	*W* (*kg*)	μ_0	μ_d
■	Steel A	1.44	0.30	1.20
□	Steel A	1.81	0.20	1.10
▼	Steel A	6.80	0.20	0.70
◇	Steel A	32.2	0.05	0.30
△	Glass A	1.81	0.20	1.10
▽	Perspex A	1.81	0.40	1.20

EFFECT OF TRACK MATERIAL AND LOAD

The nature of the track material affected the magnitude of the friction under all conditions; for instance, Perspex resulted in initial friction values almost double those with steel or glass. However, it was found that if the time-rise in static friction was expressed in terms of the initial value (μ_0) and the unlubricated value (μ_d), the results could be correlated on the same time scale, as shown in Fig. 2. Much of the scatter in this graph arises from plotting the difference of two friction values.

Similar effects were apparent with different loads; in this case changes in μ_0 and μ_d were considerable but the time-scale appeared to be unaffected over the 20-fold change in load. This finding confirms CHEYNEY's[9] experimental results using O-rings,

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but is nevertheless surprising in view of the direct influence of load in the initial approach period.

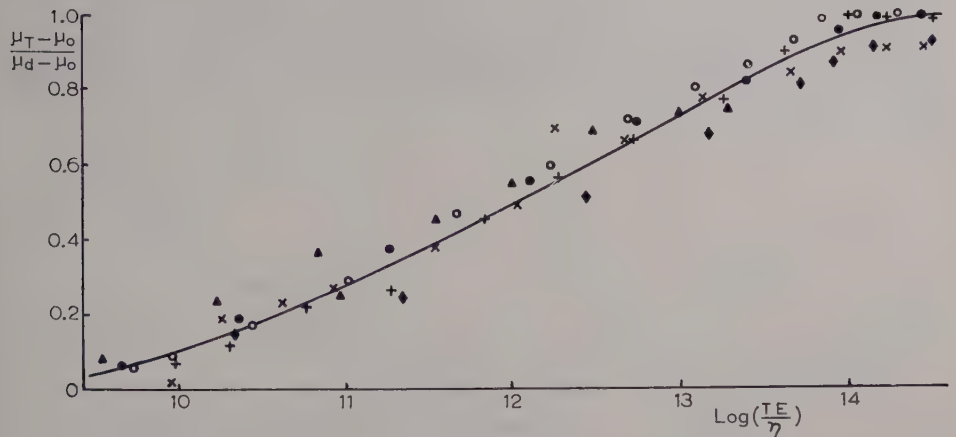

Fig. 3. Effect of lubricant and sample material. $W = 1.81$ kg; $R = 0.24$ cm.

	Lubricant	*Track*	*Sample*	μ_0	μ_d
o	Water	Perspex A	B	0.73	0.93
•	Oil E	Steel A	B	0.35	1.03
+	Oil C	Perspex A	A	0.05	0.70
×	Oil F	Steel A	A	0.15	1.10
▲	Oil B	Steel A	B	0.30	1.03
◆	Oil B	Steel A	C	0.20	1.10

EFFECT OF RUBBER MATERIAL AND LUBRICANT

Different rubbers and lubricants usually resulted in variations in the time-scale of the process in addition to changes in μ_0, and it was found that results could be correlated without unreasonable scatter on a simple basis of the viscosity of the lubricant and the elastic modulus of the rubber. In Fig. 3, results for smooth tracks are shown plotted as a function of TE/η. This empirical group, by chance non-dimensional, appears to be applicable over a 50-fold range of E and a 1000-fold range of η. The fact that the time-scale is independent of load presents no anomaly if the modulus E is looked upon as $W \cdot E/W$; for load, in addition to governing the pressure available to eject the lubricant, also governs the deformation of the flow channels. It appears to be coincidental that the two effects are of similar magnitude.

SOLID LUBRICANTS

Solid lubricants such as graphite, mica or molybdenum disulphide when used in dry form did not give rise to time effects, but on the contrary tended to reduce the friction under all conditions. When added to liquids in colloidal form, such lubricants did not influence the rise in static friction except to determine the ultimate value of friction. Results were found to follow the usual curve provided an appropriate value of μ_d was used. Results for colloidal graphite in oil are included in Fig. 3 (oil C).

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EFFECT OF TRACK ROUGHNESS

When the track surface was roughened, the static friction tended to level off at a value below the unlubricated figure. In the initial stages, however, the rise in friction was unaffected. This can be seen in Fig. 4 where friction measurements relating to one particular rubber/lubricant combination are given for several different track surfaces. In each case the samples were towed parallel to the direction of maximum surface roughness.

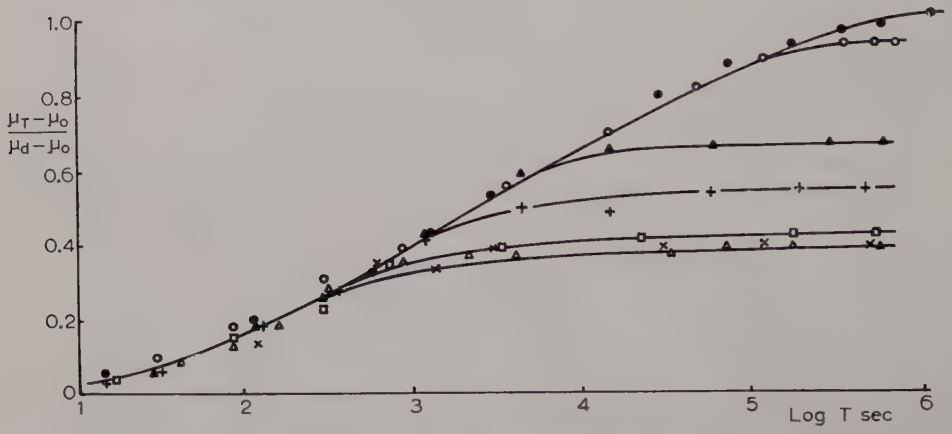

Fig. 4. Effect of track roughness. Lubricant B (Light Mineral Oil); $W = 1.81$ kg; $R = 0.24$ cm.

	Track	*Sample*	μ_0	μ_d
●	Steel A, Glass A	A	0.20	1.10
○	Steel B	A	0.30	0.90
×	Glass B	A	0.78	1.10
+	Steel C	A	0.40	0.97
△	Steel A	D (Rough)	0.23	1.17
▲	Perspex B	A	0.81	1.20
□	Steel D	A	0.60	1.03

The rise in static friction decreased as the roughness of the track increased, but it also depended greatly on the type of surface finish. For instance surfaces prepared by random abrasion or ones, such as ground glass, having no marked directional properties were found to have greater influence on the phenemenon than surfaces of similar roughness prepared by unidirectional abrasion. Fig. 5 shows that a directional surface has to be considerably rougher than a random surface in order to produce comparable effects; and it is also clear from these results that both the actual and the percentage rise in friction above the initial value, are least for the random surfaces.

In practice there would, however, be little point in deliberately using rough tracks, for, as can be seen from the values given in Fig. 4, the small rise in friction is achieved, not by limiting the ultimate friction, but by increasing the initial value.

The alternative procedure of roughening the rubber surface, was found to combine

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low initial friction with slight increase. Fig. 4 shows that with a rubber sample abraded on fine emery paper, the rise in static friction was limited to 40% of that with smooth samples. This method of reducing the ultimate friction would of course only be effective in practice where the rough surface can be maintained as the material wears. But it is found that seals and bearings made from fabric-reinforced rubber do in fact exhibit this property and consequently do not suffer greatly from time effects.

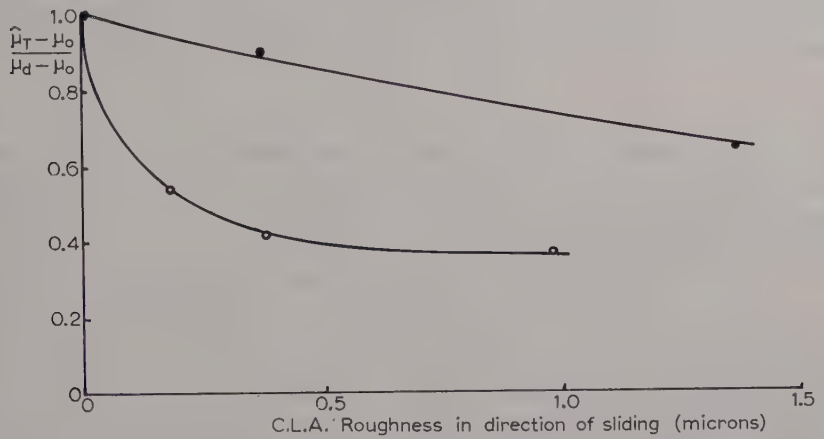

Fig. 5. Effect of surface preparation. Lubricant B (Light Mineral Oil; $W = 1.81$ kg; $R = 0.24$ cm
● Surfaces abraded in one direction; o Rough surfaces with no directional properties.

DISCUSSION

The experiments have shown that the static friction of lubricated rubber must be considered as a transient quantity, starting at zero and reaching a constant value only after a considerable period of time. All the evidence supports CLAYTON's hypothesis that the increase in static friction results from squeezing out of the lubricant film, and there is no reason to suppose that bonding or interlocking of the surfaces plays any part.

The initial period of rapid rise in friction follows the same course as with rigid surfaces, but is likely to be so short in practice that it can be ignored and the friction coefficient at the change in regime be regarded as the initial value. The second stage of increase is undoubtedly an extension of the initial stage and appears to arise because ejection of the lubricant can continue long after the surfaces have made contact over a substantial area. It can be imagined that events slow down as the lubricant film is progressively separated into numerous tiny flow channels; and the dependence of the rate of events on the modulus of the rubber tends to support this view. The results obtained with rough surfaces suggest that some finishes are capable of trapping the lubricant, and so limiting the rise in friction.

There is no evidence that the lubricant is behaving other than as a Newtonian liquid. In their study of flow of thin films between rigid flat plates MARDLES AND BIELAK[8]

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found that at an early stage in the thinning, the influence of the bulk viscosity of the lubricant disappeared, the film behaving more as a plastic solid. The absence of such effects in the present experiments is probably due to the use of loads some 1000 times as great as in MARDLES' work, so that rheological effects were insignificant.

That the ultimate static friction tended to approach the unlubricated value suggests that, even after cleaning, the rubber surfaces remained contaminated by oily or waxy plasticisers used in the mixing process. With unfilled rubbers the friction would probably never reach that corresponding to unlubricated sliding; but this has not been verified.

ACKNOWLEDGEMENT

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STATISTICS ON CYLINDER WEAR IN MARINE DIESEL ENGINES* **

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SUMMARY

A description of the United Steamship Company's efforts to elucidate statistically the problems in connection with cylinder wear in marine diesel engines, by the use of punched cards and electronic digital computers. Further, some hypotheses concerning the causes of cylinder wear and the way to overcome it are given.

ZUSAMMENFASSUNG

STATISTISCHE BEHANDLUNG DES ZYLINDER-VERSCHLEISSES IN SCHIFFS-DIESELMOTOREN

Die United Steamship Company (D.F.D.S.) versucht das Problem des Zylinderverschleisses von Schiffs-Dieselmotoren durch Anwendung statistischer Methoden aufzuklären, wobei Lochkarten und elektronische Rechenmaschinen gebraucht werden. Dieses Projekt wird beschrieben. Einige Hypothesen über die Ursache von Zylinderverschleiss und Mittel, um diese Ursachen auszuschalten werden ebenfalls mitgeteilt.

The immense economic importance which the replacement of cylinder liners means to shipping — in particular since heavy fuel has come to be used so extensively for the operation — has resulted, in the course of time, in numerous lectures, theses and articles on the causes of cylinder wear.

The fact that, despite the valuable work carried out by experts throughout the world, no clear picture of the conditions of wear is available so far must be considered to be due to the large number of variables which must be taken into account.

Even if consideration is confined to one definite type of engine, the number of variants is so vast that it must be regarded as being impracticable to handle without the help of a machine.

In the following will be shown how The United Steamship Company uses mechanical

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means to record the conditions which are believed to exert an influence on cylinder wear, and the advantages that are obtained thereby.

For elucidation of the conditions of wear, a current collection of details from motor vessels has been started. These details are handled first individually from different points of view; thereby ideas are obtained for the formation of hypotheses, and finally an attempt is made to verify these hypotheses statistically.

In order to be able to handle the material collected statistically it is necessary to produce a method of registration which in an easy and convenient manner allows the material to be considered from different angles and to be divided into various subgroups. For this purpose the material is registered on punched cards.

The punched card may be considered to be a free link in a card index, and has 3 functions:

(1) The punched cards can be mechanically sorted and arranged in different card indexes or systems as required.

(2) They can give their numerical information to a tabulator; the latter is a combined calculating machine and typewriter. The tabulator can carry out comparatively simple, but from the point of view of work rather extensive, calculating tasks..

(3) Finally, the punched cards can be read directly into the electronic calculating machine DASK*, which can carry out the actual, complicated statistical computations in a very short time, even when very extensive and very complicated calculations are involved, such as regression analyses and correlation computations.

With the help of statistics and extensive information it will probably be possible to confirm or invalidate the theories that have appeared or may appear concerning the factors which enter into different combinations in connection with cylinder wear.

METHOD

(a) Data

The most essential details that are reported by the ships following each inspection of pistons are the following:

Cylinders: the total number of working hours of the cylinders, the number of working hours since the last inspection, the wear in the fore-and-aft direction and in the transverse direction at the top, 200 mm from the top, in the middle and at the bottom.

Pistons: total number of working hours, total number of working hours since the last inspection, the total number of piston and scraper rings replaced on account of wear or breakage.

Fuel valves: working hours, opening pressure after removal, pressure during spraying, and finally an ordinary estimation of the condition of the valve at the time of replacement.

Working conditions: number of revolutions, temperature of cooling oil and cooling water, scavenging air pressure and temperature, P_i (indicated mean pressure), P_{max} and P_k (compression pressure), exhaust temperature as well as the consumption of cylinder oil.

* Dansk Sekvens Kalkulator.

Finally, in connection with each *liner*, when so indicated by the manufacturers, information is available about Brinell hardness, cooling time, inoculation, the analysis proper and, finally, the pig iron composition.

By registration of these data the most essential factors affecting the cylinder wear are considered to have been obtained.

In the normal course of affairs the registered material is applied in the following way: every three months a statement is received from the Punched Card Department of the latest information regarding cylinders and pistons.

(b) Punched cards

The sorting machine sorts the punched cards, first according to ships, next according to cylinders, and within each range of numbers of cylinders according to dates. The Technical Department thus has a complete check on the immediate state of cylinders and pistons in all the motor-vessels of the fleet. Having an up-to-date card index affords very substantial advantages, not the least of which is in connection with the purchase of spares and the planning of annual dry dockings.

In addition, various hypotheses can be tested in various ways. Sorting may be carried out according to make, Brinell hardness, whether the cylinders are operating on diesel oil, mixed oil or fuel oil, whether the material contains vanadium or titanium — on the whole according to all possible criteria.

To give an idea of the simple and rapid manner in which this work can be performed, it may be mentioned that the sorting machine can handle 42,000 cards per hour, and the tabulator can write 9000 lines per hour. The three-monthly survey, which at present involves approximately 2000 cards, can be written in fifteen minutes. During the two years in which the system has been in use, the punched cards machinery has been occupied with this task for a total of 62 hours, and it is anticipated that the future load, when the first teething troubles will have been overcome, will be 20–25 hours a year. The unit of work that is required for controlling, registering, codifying and punching the information involved is 1 man-hour per day, and in view of the immense economic importance which cylinder wear means to shipping, this is regarded as labour well spent.

The material which up to now has been treated statistically is information about cylinder liners in diesel engines of Burmeister & Wain's type DM 50-VF-90, a 2-stroke trunk engine with 500 mm piston diameter and 900 mm stroke. 33 ships are powered by this type of diesel engine. Altogether 292 cylinders are in operation. 438 liners have been treated, and the number of observations from these liners amounts to a total of 2119. Analyses of 336 liners are available.

In order to be able to make comparisons between the individual liners it has proved necessary to subdivide the ships into groups (Table I). The load of the engines — in this connection, medium pressure and number of revolutions — the plying limits of ships, the number of hours they are operating consecutively, whether there are frequent manoeuvres, whether fuel oil, mixed oil or diesel oil is used for the operation;

all these factors influence essentially the cylinder wear and necessitate subdivision of the ships into groups.

TABLE I

PUNCHED CARD MATERIAL CONCERNING CYLINDER LINERS IN ENGINES OF THE TYPE DM 50-VF-90 ON 26TH MARCH, 1958

	Group	Number of ships	Number of cylinders	Number of liners	Number of cards	Number of analyses
01	England boats	2	40	49	264	
03+ 04	Coastal trade vessels	4	55	83	315	
07	Exports boats	6	36	52	231	
18	North America boats	4	32	59	307	
20	Smaller North America boats	3	22	42	210	
22	Mediterranean boats	6	48	69	362	
24	B-boats	6	36	55	332	
020+040	Oslo boats	2	23	29	98	
	Total	33	292	438	2119	336

Within each group of ships an attempt has been made subsequently to arrive at a "normal wearing curve" by means of figures so as to provide a control. If, for example the "normal curve" is produced once every six months, a survey is afforded as to whether conditions of wear within each individual group of ships have improved or deteriorated, and further a good scale is obtained for the effect of purposely introduced changes in operation conditions.

(c) Statistical treatment of data

The task is thus to trace the progress of wear for a normal liner on the basis of the given measurements, so that we gain a picture of more than just the individual properties of the linings involved, originating from make, hardness, alloy, etc. Endeavours in this respect, however, can be successful only if the various properties of the liners are represented equally within the different groups of ships and numbers of hours.

In this respect the completeness of the information still leaves something to be desired; subject to reasonable reservations, however, it can be used.

The detected mean values of the wear within each 1000 hours form the background for a leveling, which will then be the basis for the theoretical function showing the dependence of the wear on the number of working hours within each group of ships. This is based on the assumption that these mean values are reasonably equivalent representatives for individual observations within each period. This means that it is necessary to investigate the deviation of the individual observations from the mean values, or to calculate their standard deviation from the mean values.

The standard deviations serve to illustrate accumulation around the mean values. Thus little deviation means substantial accumulation and consequently greater reliabity of results, whereas in the case of large deviation and subsequent small accumulation greater care must be exercised in using the mean values for comparison, and in drawing conclusions.

The results obtained show somewhat variable standard deviation, which indicates that the mean values are subject, in this connection, to certain arbitrary fluctuations probably originating from different representations of make and analysis, as well as possible errors in measuring.

By simple levelling — here we have *a sliding* average — we obtain a so-called progress curve or "trend" showing the wear as a mathematical function of the number of working hours. The levelled curve can be computed only within a certain period of working hours, and in order to obtain a complete curve a function has been calculated which expresses the dependence of wear on the number of hours, and when represented graphically coincides as closely as possible with the levelling curve, and can then with some justification be considered a "normal curve of wear".

The function is computed as a third degree polynomium:

$$Y = ax^3 + bx^2 + cx + d$$

where Y is the wear and x the number of 1000 hours. Thus a function has been constructed which by integration can give a unique expression for the total wear within a definite period of working hours. At this juncture we had to decide whether to apply this form of computation, or to perform a regression analysis, where the wear is the dependent variable factor, the number of hours being the independent variable factor. By linear regression, in a system of co-ordination, a straight line would be obtained expressing the same proportion of dependence and being so located that the sum of the square of the deviations of the individual observations from the line became the least possible. By viewing the curves for all groups of ships it can be seen that there will be no question of a linear proportion between the wear and number of hours with the material that is available at present.

By non-linear regression we should obtain a curve located in the plan following the same principle as the line in linear regression; however, such an analysis is essentially more complicated and demands more requirements from the material than this can fulfil so far. For this reason the aforementioned form of computation has been chosen.

Computations of "normal wearing curves" have been carried out for 3 groups of ships, *viz.* 01, 24 and 18, on the basis of observations collected up to November, 1957.

Now that the calculating centre possesses facilities for feeding information direct to the electronic calculating machine from punched cards, we have, with observations up to May, 1958, only a few days before this paper was completed, arranged for further computations to be made on a very much larger scale. The results obtained thereby have not yet been considered thoroughly, but for the above-mentioned 3 groups of ships a comparison can be shown between 2 sets of normal curves for a six-monthly period and thereby these curves can be used as a control.

The computations performed by DASK comprise all 8 groups of ships (a total of 1864 measurements), and for each group the following has been calculated:

(1) Mean wear for each number of 1000 working hours.
(2) Standard deviation from each mean wear.
(3) Sliding average for each mean wear.
(4) The constants a, b, c, d in the "normal curve"

$$Y = ax^3 + bx^2 + cx + d.$$

This has been carried out for the wear

at the top of the liners in the transverse direction,
at the top of the liner in the fore-and-aft direction,
200 mm from the top in the transverse direction, and
200 mm from the top in the fore-and-aft direction,

so that 4 normal curves per group of ships are obtained.

(5) Finally, for each observation the proportion is calculated between the wear in the transverse direction and the wear in the fore-and-aft direction, partly at the top and partly at 200 mm from the top, and also for each number of 1000 hours, the mean values and standard deviation of these proportions. This was made with a view to testing the ovalness of the cylinder liners.

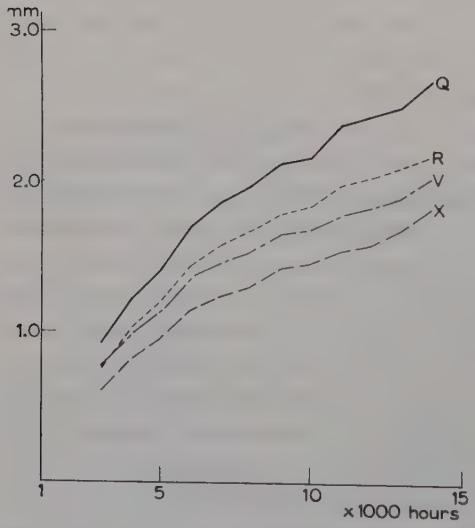

Fig. 1. Group 01. Sliding average of cylinder wear.

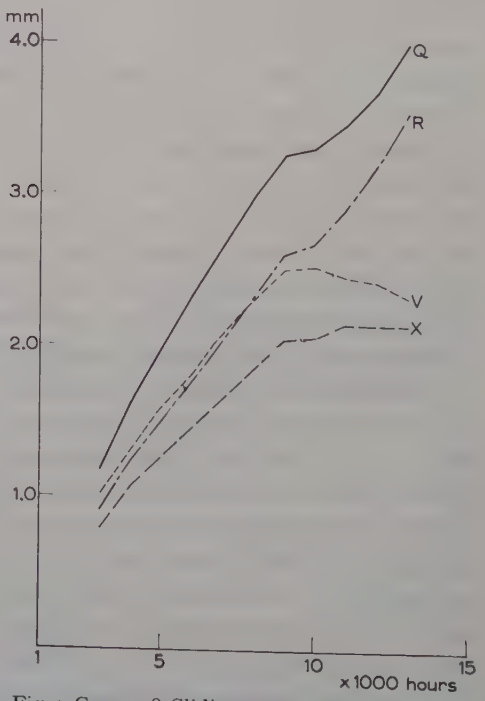

Fig. 2. Group 18. Sliding average of cylinder wear.

RESULTS

In Figs. 1, 2 and 3, the sliding average is shown graphically for the groups of ships 01, 18 and 24. Curve Q shows the wear at the top in the transverse direction, curve R shows the wear at 200 mm from the top in the transverse direction, curve V shows the wear at the top in the fore-and-aft direction, and curve X shows the wear at 200 mm from the top in the fore-and-aft direction.

Figs. 4, 5 and 6 show the normal curves (the solid line curves) and the sliding average (the dotted curve) for the wear at the top of the liners in the transverse directions, partly from November, 1957, and partly from May, 1958. It can be seen that the course of the normal curves falls very much in line with the course of the sliding average over the period from 3,000 to 14,000 hours.

In order to arrive at an expression for the total wear, the "normal curve" is integrated from $x = 3$ to $x = 14$. The area calculated thereby between the normal curve and the axis of abscissa is a unique expression for the total wear, and after this, comparisons can be made for the 3 groups of ships, between the wear assessed in November, 1957, and that assessed in May, 1958. Table II shows the result of this integration. The normal curves thus give a picture of the wearing effect for the different operational conditions of each group of ships, and further the comparison between the survey in November, 1957, and that of May, 1958, shows the effect brought about by purposely introduced alterations for the groups 18 and 24. These alterations will be discussed later on in this article.

TABLE II

Group	*Normal curve of wear.* y = wear in 1ooths of mm; x = 1000 hours	
	November, 1957	*May, 1958*
01	$y = 0.086x^3 - 2.870x^2 + 39.868x + 20.918$	$y = 0.234x^3 - 6.852x^2 + 75.610x - 87.784$
24	$y = 0.062x^3 - 1.971x^2 + 41.248x + 6.576$	$y = 0.031x^3 - 1.374x^2 + 39.564x - 30.256$
18	$y = 0.146x^3 - 4.190x^2 + 65.366x + 0.670$	$y = 0.290x^3 - 8.256x^2 + 98.179x - 117.180$

Group	*Calculated areas in 1ooths of mm² 3,000 to 14,000 hours*				
	November, 1957		*May, 1958*		
	Area	*Index*	*Area*	*Index*	*Index Nov., 1957 = 100*
01	2183	100	2141	100	98
24	2738	125	2419	113	88
18	3724	171	3193	149	86

An attempt has been made to test in a simple statistical manner the influence of the Brinell hardness on wear. Within each group of ships the cards are sorted into Brinell hardness ranges. Fig. 7 shows the normal curve for group 18; the Brinell hardness range curve is close to the normal curve in all the Brinell hardness ranges, and the picture is alike for all the groups. The curves, incidentally, illustrate clearly the lack of completeness of the material.

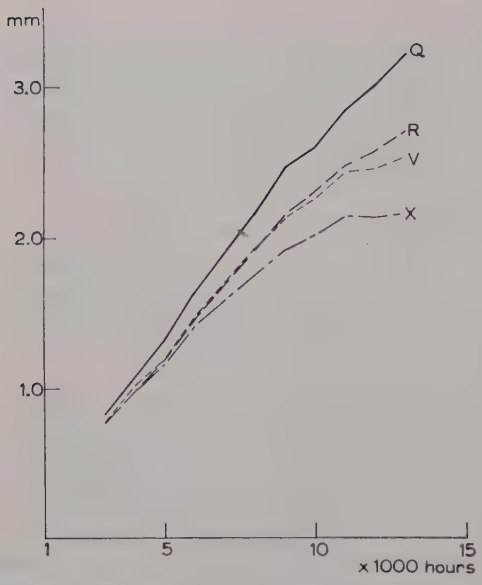

Fig. 3. Group 24. Sliding average of cylinder wear.

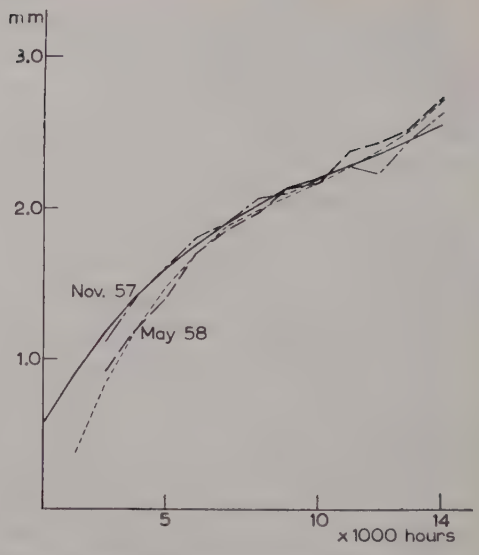

Fig. 4. Group 01. Normal curves of cylinder wear.

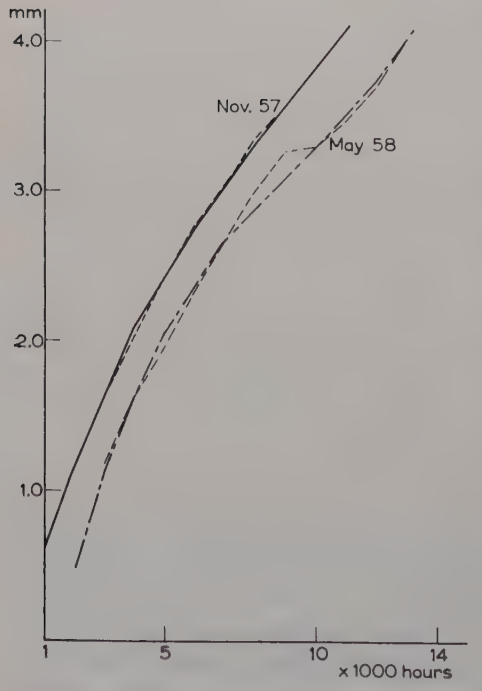

Fig. 5. Group 18. Normal curves of cylinder wear.

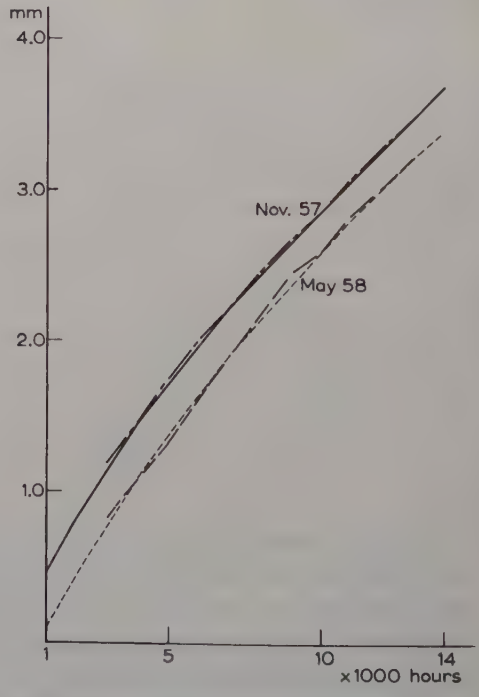

Fig. 6. Group 24. Normal curves of cylinder wear.

If we now count within each Brinell hardness range for each group how many observations are above and below the normal curve within each period of time, it will be found that the observations for all the groups within all the ranges are evenly distributed, and it can thus be concluded that there is a certain probability that the

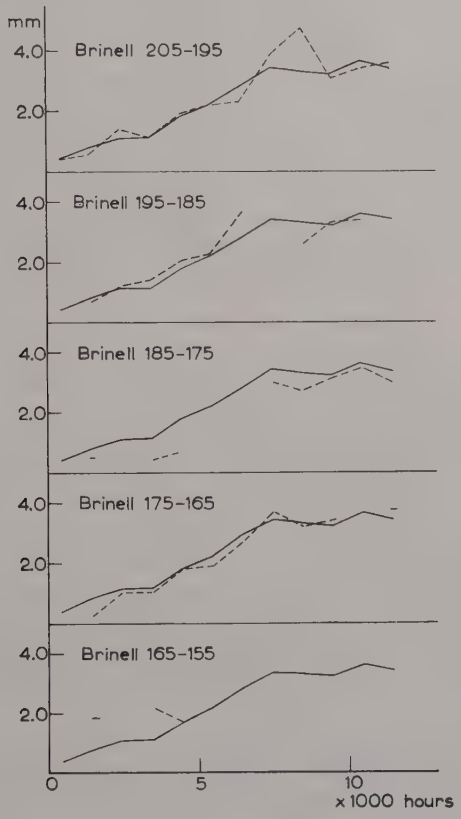

Fig. 7. Group 18. The influence of the Brinell hardness on the cylinder wear.

Brinell hardness within rather wide limits has no essential influence upon the wear. However, it must be admitted that the examination has the drawback that the Brinell hardness of the piston rings is not included in the examination. The reason for this is, to avoid the necessity of additional record-keeping by the chief engineers in maintaining a log-book of the hardness of the piston rings, but considering that the rings are for the most part of the same make, and random checks of the rings having shown very small variations in the Brinell hardness, it is considered reasonable to draw the above conclusion.

With a view to future purchases, tests have been made to ascertain how the liners of various makes behave as regards wear. For each group of ships the wearing results of the respective makes were illustrated. The distance in per cent from these points to the normal curve was shown graphically for each group of ships and for each make.

Fig. 8 shows how the curves according to make appear for groups 3 and 4. It will be seen that make No. 5 is satisfactory, make No. 4 is poor, whereas 1, 2 and 3 are almost on the same level. In order to obtain a general estimation, 3 mutually independent persons have inspected these curves for all the groups and given marks; these have been summed up. The evaluations were in close agreement, and consequently

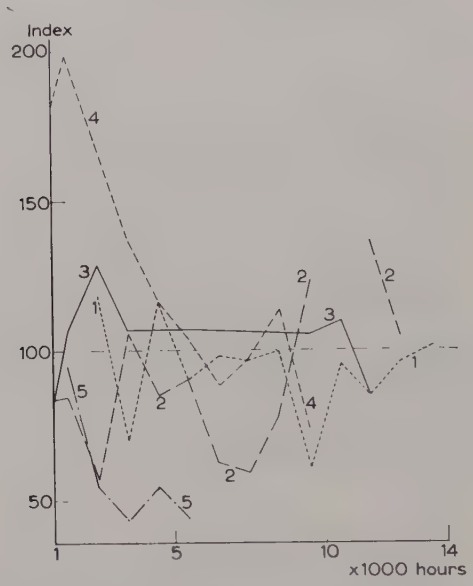

Fig. 8. Group 03 + 04. The wearing results of various makes.

there is reason to assume that by this method a good assessment is obtained for the various makes.

An attempt has been made to arrive at a fuller recognition of the value of the analysis in assessing the wearing properties. Sorting has been done and comparisons have been made using all possible criteria, but the available material is not sufficient to enable reliable conclusions to be drawn, observations being too scattered for this purpose.

One thing, however, is thought to have been ascertained with some certainty, *viz.* that the general analysis normally furnished by the manufacturers is not sufficient in all cases to characterize the wearing properties of a cylinder liner. The bases for this conclusion are 31 liners of the same make; the rate of wear in all of them was approximately twice the normal rate. Analyses and grindings have been made of these liners, but neither the analysis nor the grinding could give any explanation of the essential difference in the wearing properties.

As the result of this experience, attempts are being made to extend the analysing cards.

Some typical examples

In the following the conditions of wear in a number of ships are shown graphically. As abscissa, the working period has been set off with 1000 hours and as ordinate the maximum wear in millimetres.

Fig. 9. M/S *"Paraguay"*, one 8-cylinder, 2-stroke trunk engine of the B. & W. type DM 50-VF-90. During the first 28,000 hours the engine ran on pure diesel oil; then it was changed to a mixed oil, 50 % fuel oil + 50 % diesel oil. Since most of the cylinders had been in operation for relatively many hours before the change over to operation on mixed oil, no essential aggravation of the wear was observed after the change to mixed oil. The cylinder had had time to work up a level. If, on the other hand, a new cylinder is run-in on mixed oil, we notice a highly prolonged running-in wear. After approx. 55,000 hours the engine was provided with cylinders bored to oversize, and a very fine running-in wear was recorded. This condition will be dealt with later.

Fig. 10. M/S *"Ficaria"*, one 8-cylinder, 2-stroke trunk engine of the B. & W. type DM 50-VF-90. After 4000 hours running a change to mixed oil was made. Here the cylinders had not had time to build up any level; essentially increased wear has been observed. The first two cylinders, used as cylinder 5, are of the above-mentioned make, the wear of which was unusually severe. The third cylinder, which is installed as cylinder 5, shows a normal rate of wear.

Fig. 11. M/S *"Lemnos"*, one 8-cylinder, 2-stroke trunk engine of the B. & W. type DM 50-VF-90. After approx. 9000 hours a change to mixed oil was made. The special feature about this ship is that the first four cylinders are lubricated with a mixture of ordinary mineral straight cylinder oil with the addition of "colloidal" graphite. The improved wear conditions obtained are negligible.

The consumption of piston rings in the first four cylinders, however, is somewhat lower than in the remaining part of the engine.

Fig. 12. M/S *"Colombia"*, one 7-cylinder, 2-stroke crosshead engine of the B. & W. type DM 62-VTF-115. After 16,000 hours' operation a change was made to mixed oil, and at approx. 37,000 hours' operation an additivated emulsion cylinder oil was introduced; this gave an essential improvement in the wearing results. The wear is now 47 % of the rate recorded when originally operating on diesel oil.

Fig. 13. M/S *"Brasilien"* one 8-cylinder, 2-stroke crosshead engine of the B. & W. type DM 62-VFT-115. After 5000 hours a change was made to heavy fuel oil, and later on an additivated emulsion oil was introduced as cylinder oil; in this case also, it caused a reduction in wear.

Fig. 14. M/S *"Ohio"*, one 6-cylinder, 2-stroke crosshead engine, turbo-charged, of the B. & W. type DM 50-VTBF-110. During the first 5000 hours diesel oil was used. The cylinder oil is an additivated, emulsified oil. The wearing result is unusually good for the 5 cylinders; only cylinder No. 5 shows an abnormal rate of wear.

Fig. 15. M/S *"Oklahoma"*, one 6-cylinder crosshead engine (sister-ship to M/S *"Ohio"*). Diesel oil is used for operating, and as cylinder oil a non-emulsified, but

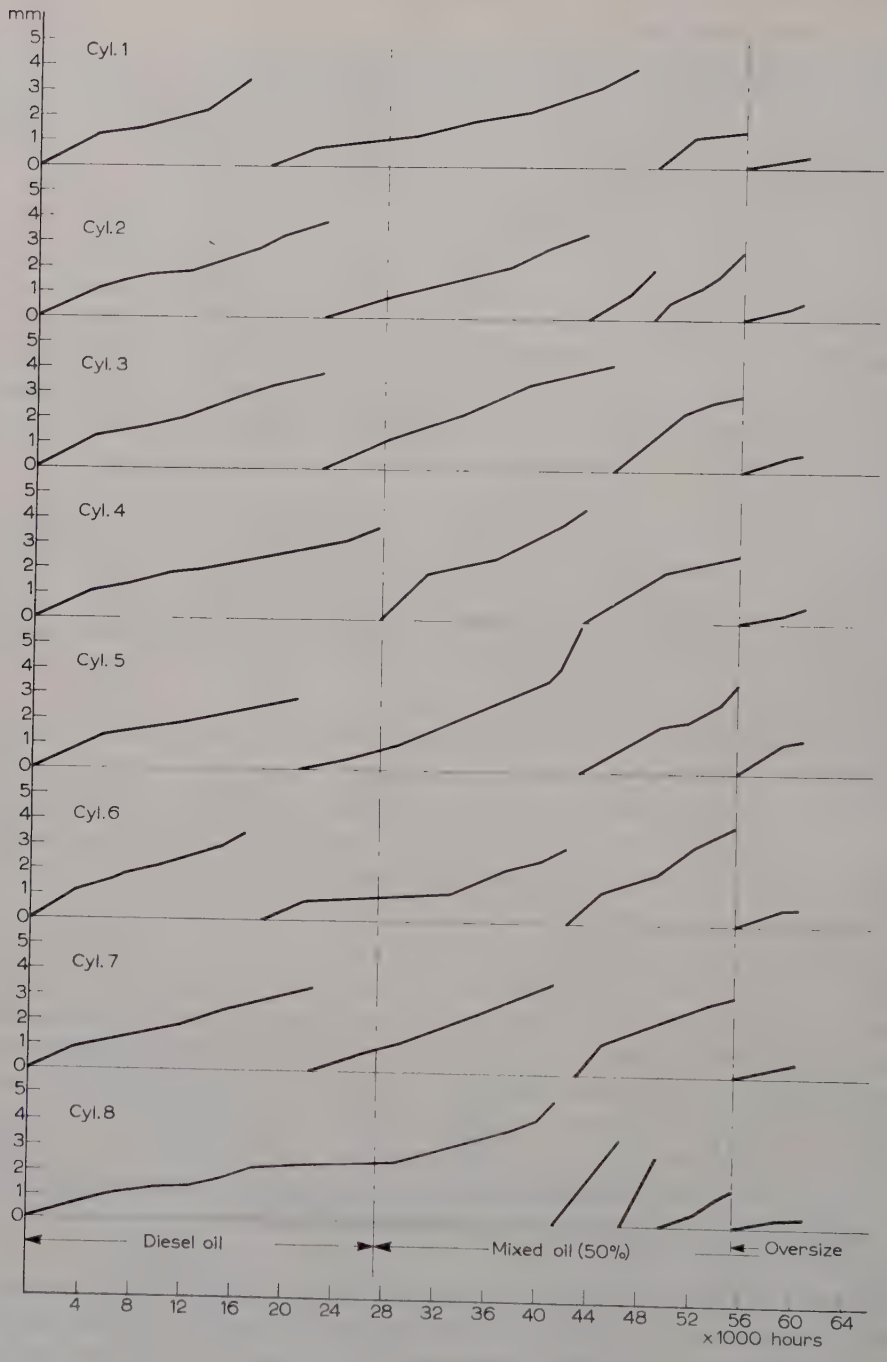

Fig. 9. Cylinder wear. M/S "*Paraguay*". DM 850-VF-90. Cylinder oil: Tromar 85.

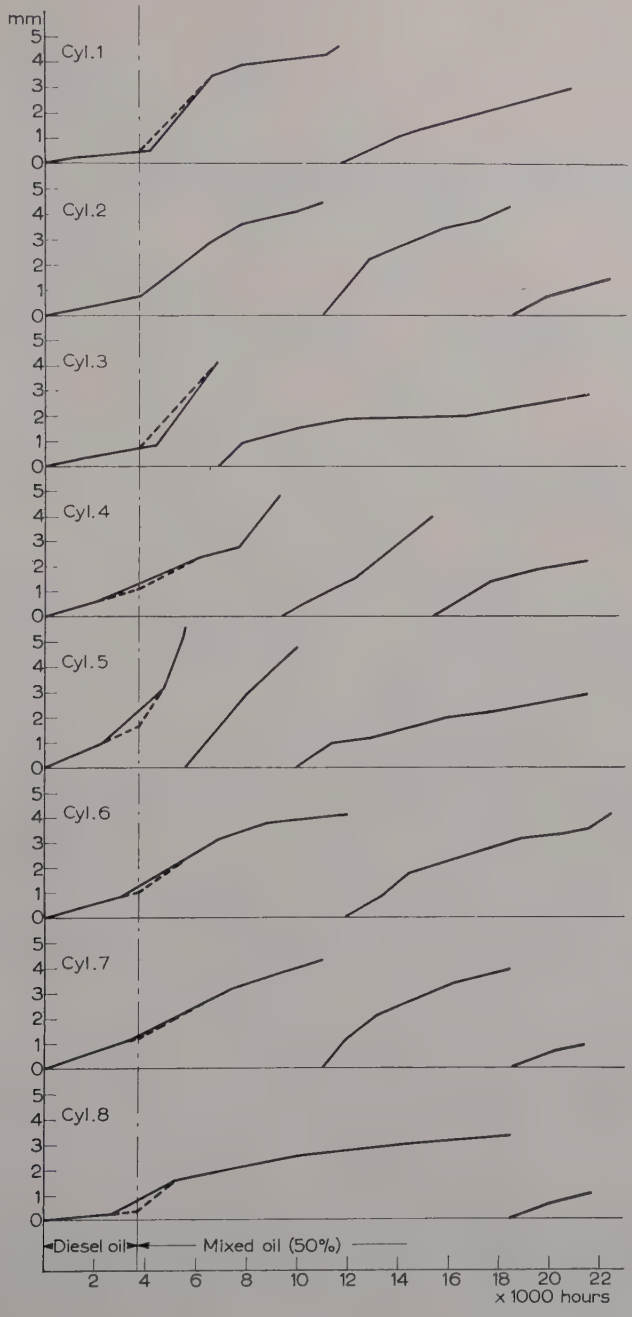

Fig. 10. Cylinder wear. M/S "*Ficaria*". DM 850-VF-90. Cylinder oil: Ilo cylinder oil SR.

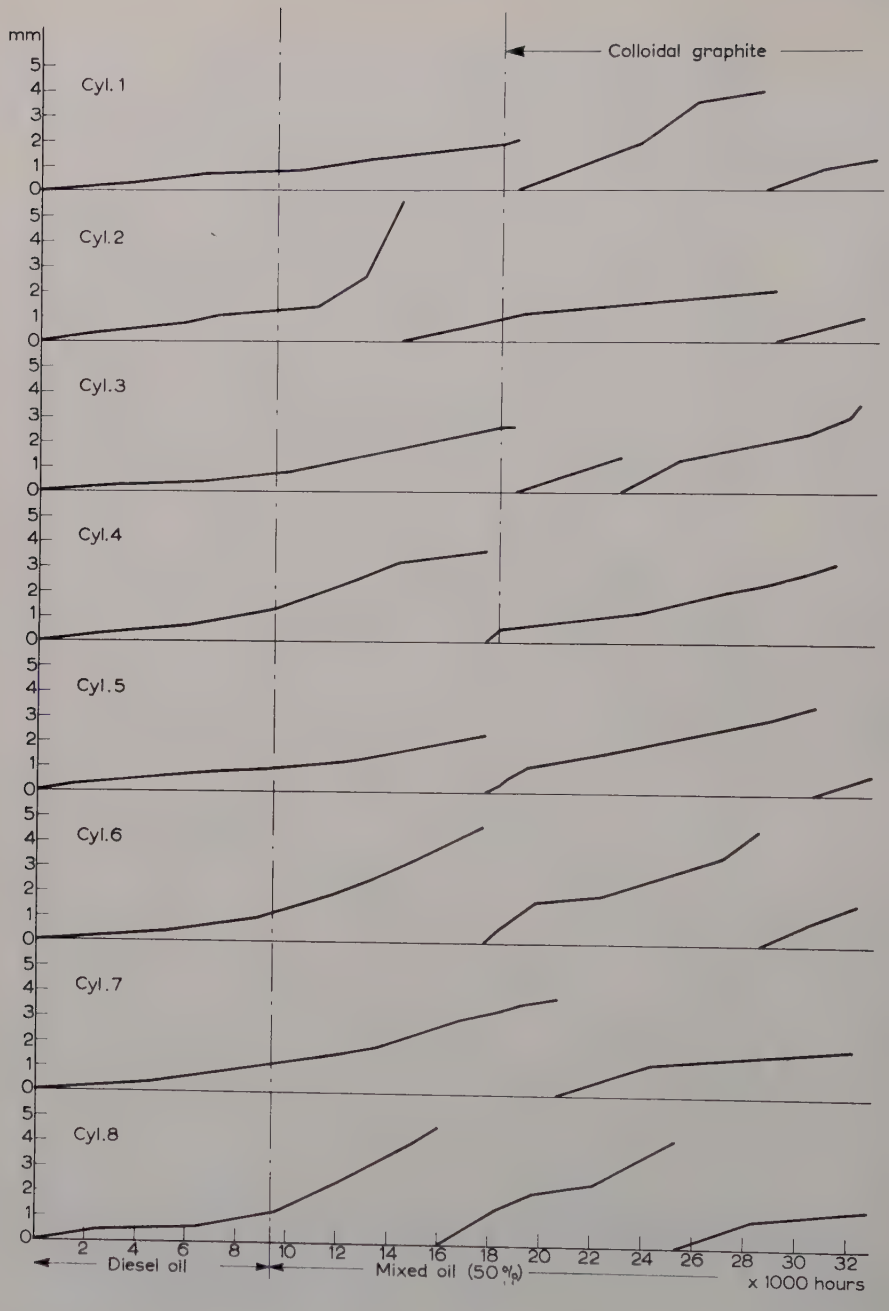

Fig. 11. Cylinder wear. M/S "*Lemnos*". DM 850-VF-90. Cylinder oil: Deusol C.

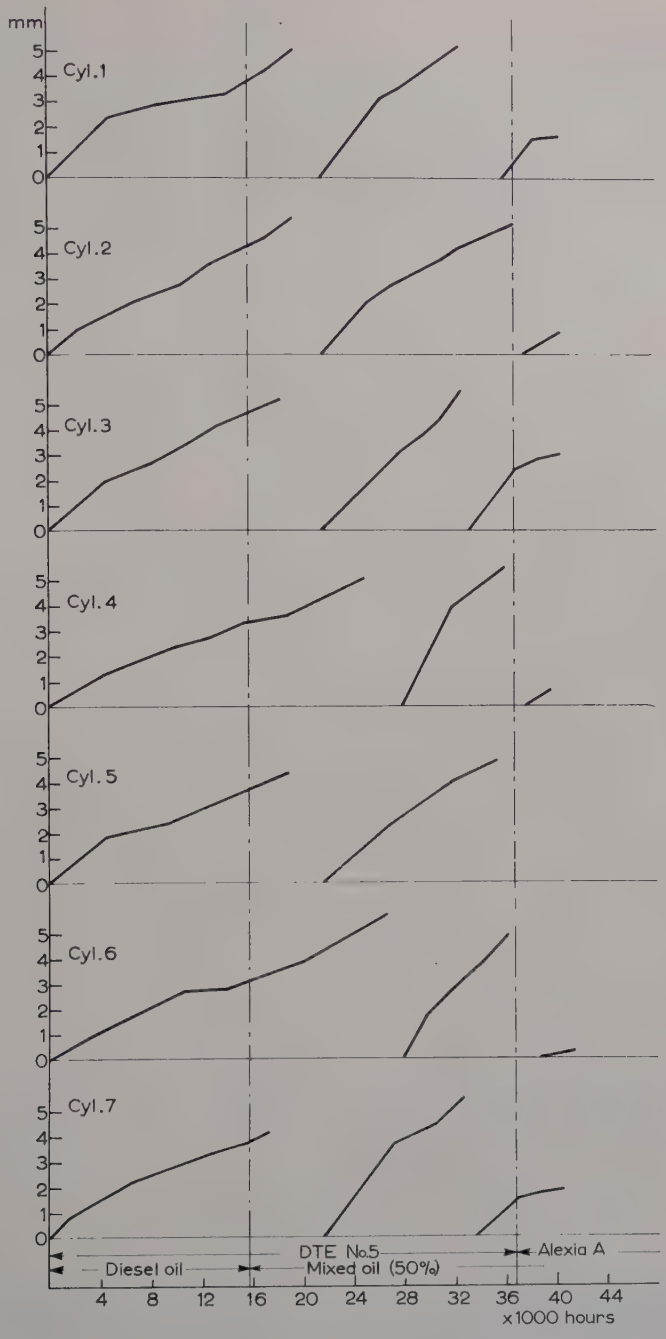

Fig. 12. Cylinder wear. M/S "*Colombia*". DM 762-VTF-115.

additivated cylinder oil is used. The wear is strikingly moderate for the 5 cylinders; only cylinder No. 3 shows abnormal wear.

The material from which these two abnormal cylinders are made has been analysed, and grinding has been carried out; however, neither the analysis nor the grinding could provide any explanation of the abnormal conditions of wear.

Fig. 16. M/S *"Prinsesse Margrethe"*, two 8-cylinder, 2-stroke turbo-charged trunk engines of the B. & W. type DM 50-VBF-90. The vessel is operating on diesel oil and an ordinary straight mineral cylinder oil. If we compare these wearing results

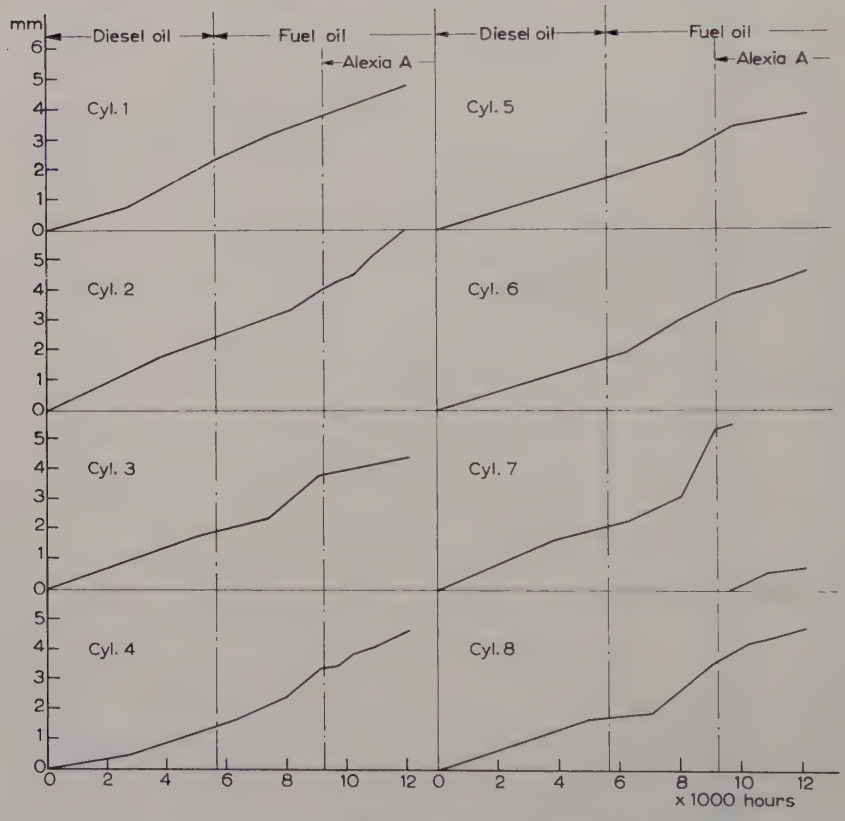

Fig. 13. Cylinder wear. M/S *"Brasilien"*. DM 862-VTF-115.

with those of M/S *"Jens Bang"*, M/S *"Kronprins Frederik"* and M/S *"H. P. Prior"* (the 3 last-mentioned ships are all provided with trunk engines of the type DM 50-VF-90) under the same conditions of operation, an essential improvement in the wearing results will be noticed, probably resulting from the greater quantity of surplus air and cooled scavenging air in the turbo-charged engines.

Reducing wear

What hypotheses do these examples lead to? Concerning the influence of fuel oil on cylinder wear, the following conclusions are arrived at: increased viscosity and higher sulphur content cause increased wear, corrosive wear being the predominant wearing factor. Under the same conditions an increased sulphur content in the oil, will undoubtedly increase the deposits of sulphuric acid on the cylinder walls, and

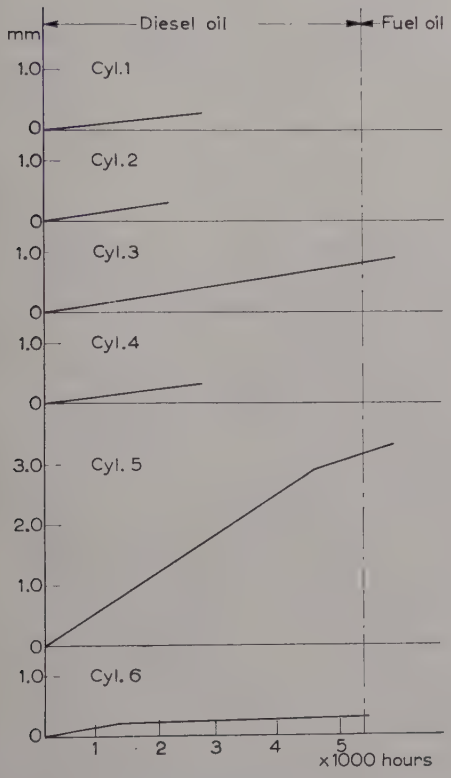

Fig. 14. Cylinder wear. M/S *"Ohio"*,, DM 650-VTBF-110. Cylinder oil: Alexia A.

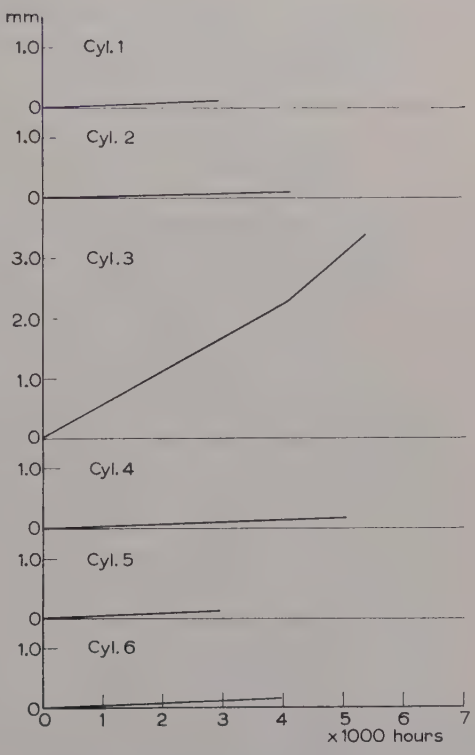

Fig. 15. Cylinder wear. M/S *"Oklahoma"*. DM 650-VTBF-110. Fuel: Diesel oil. Cylinder oil: Tromar DX 130.

the greater viscosity makes it more difficult to obtain a satisfactory atomization of the oil, even if an attempt is made by heating to provide the same viscosity at the nozzle. An essential part of the difference for the normal curves for the groups 01, 18 and 24 is probably due to this condition. Group 01 is operating on the lightest, group 18 on the heaviest fuel oil.

Simple and inexpensive measures, however, are thought to be capable of reducing

the wear when operating with fuel oil, *e.g.* it is thought to be of importance to change to diesel oil approximately 1 hour before the vessel touches alongside the quay, so that the deposits of sulphuric acid on the cylinder walls will be diminished when the latter get cold. Then, after having been at rest for about half an hour, the engines have to be turned by means of a turning-gear, extra cylinder oil being supplied at the same time from the lubricating devices. This practice, which has been introduced for groups 18 and 24, is believed to be one of the reasons for the improvements in the normal wearing curves from November, 1957, to May, 1958.

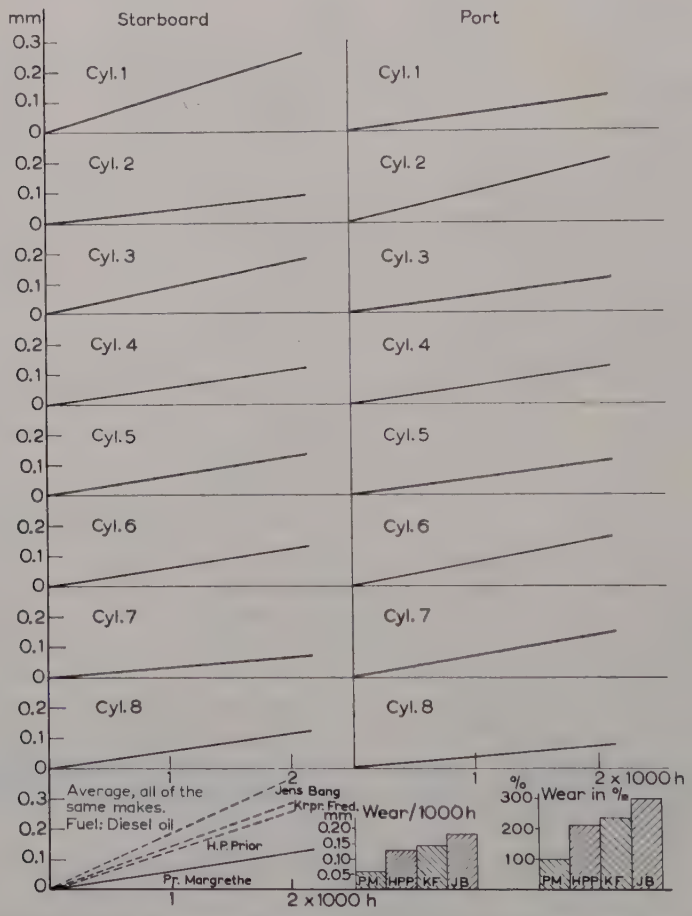

Fig. 16. Cylinder wear. M/S "*Prinsesse Margrethe*". Two DM 850-VBF-90. Cylinder oil: DTE Nr. 5.

An alternative measure for reducing corrosive wear consists of supplying anti-corrosion additives to the cylinder-lubricating medium. This procedure, however, is more suitable for crosshead engines, because the excess of cylinder oil, which is scraped down in the system oil in trunk engines, may give rise to difficulties. In certain cases the

system oil shows a sharply increasing ash content, and on using a special additive deleterious formations of foam have appeared in the system oil.

A further measure, which is thought to be of importance, is to run the engine with as hot a cooling water inlet and as cold a cooling water outlet as possible — in other words: to heat the lower part of the cylinder and cool the upper part of the cylinder as much as possible. This, however, entails rather extensive constructional alterations, which we are somewhat reluctant to undertake at present. This hypothesis has been affirmed by charting distribution of wear all over the cylinder surface in the case of 3 worn cylinders taken at random.

The cylinders were clamped in a turning lathe, centred, and the wear along the cylinder surface was then charted. In all three cases practically the same picture was obtained; this can hardly be considered as due to chance (Fig. 17). The maximum

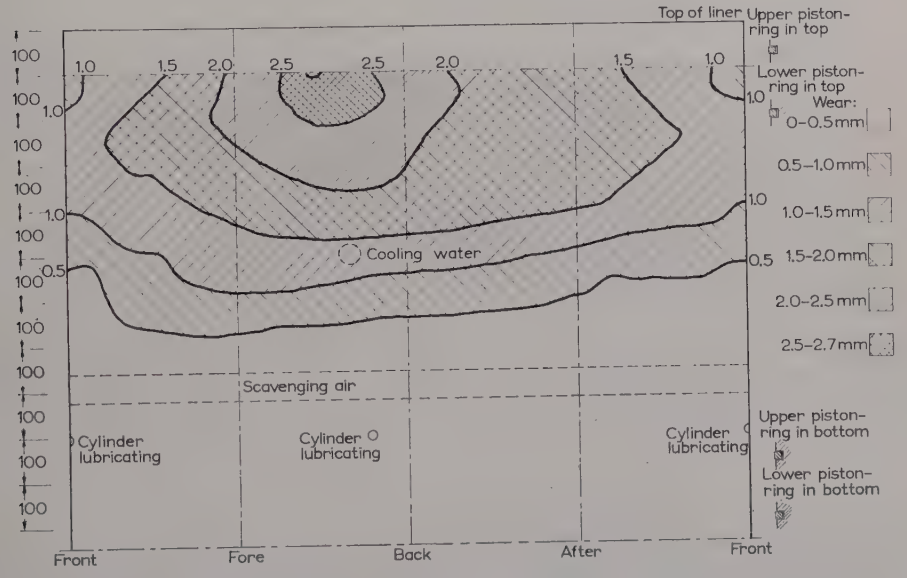

Fig. 17. Cylinder wear. M/S "*Andros*". DM 50-VF-90. Cylinder 3. June 1954-Dec. 1957 = 13,065 hours. Fuel oil: Diesel oil until October 1955 for 4,564 hours. Thereafter, heavy fuel for 8,501 hours.

wear in all cases is located at the back of the cylinder directly above the cooling water inlet, despite the fact that in this location cylinder-lubricating medium is supplied. The heaviest deposition of sulphuric acid is undoubtedly directly at the cooling water inlet, the coldest place on the cylinder, where the cylinder surface, however, has such a temperature that the oil film acts as a protective agent against corrosion attack. The sulphuric acid is then carried by the piston rings higher up in the cylinder where the temperature breaks down the oil film so that it loses its protective effect; thus at this position the degree of wear is highest. Heating the cooling water inlet would reduce the condensation of sulphuric acid and consequently the wear.

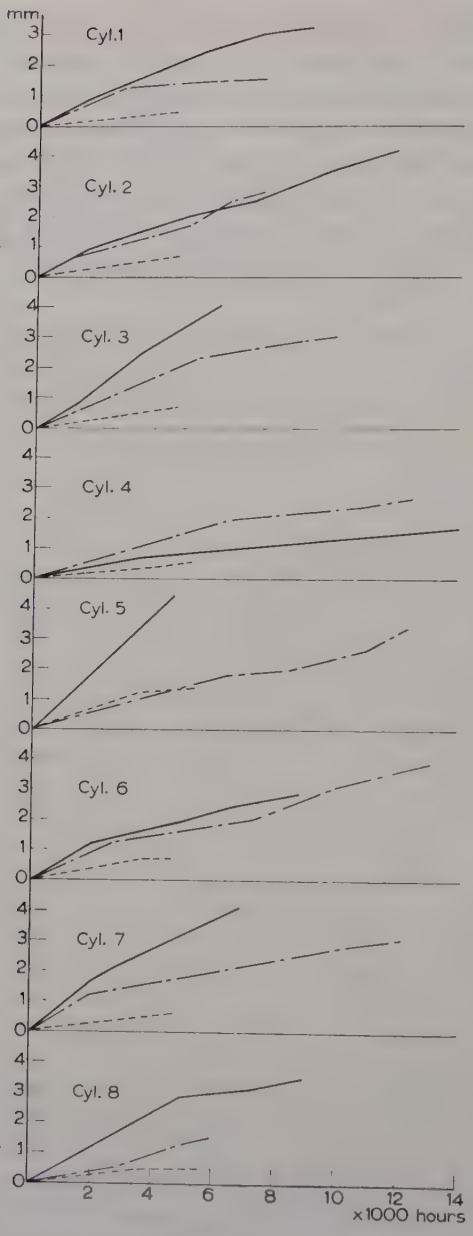

Fig. 18. Cylinder wear. M/S "*Paraguay*".

Finally, it is thought to be of decisive importance to operate with scavenging air as cool as possible; this is indicated by the wearing results for turbo-loaded engines. To add weight to this assertion, the wearing results from M/S "*Paraguay*" after the engine of this vessel had been provided with oversized cylinder liners, *i.e.* worn liners

bored to 6-½ mm, are shown in Figs. 18 and 19. In addition to the installation of the oversized liners, a cooler was inserted between the scavenging air blower and the receiver, so that under normal conditions, on an average, a cooling of the scavenging air of 25°–30°C is obtained.

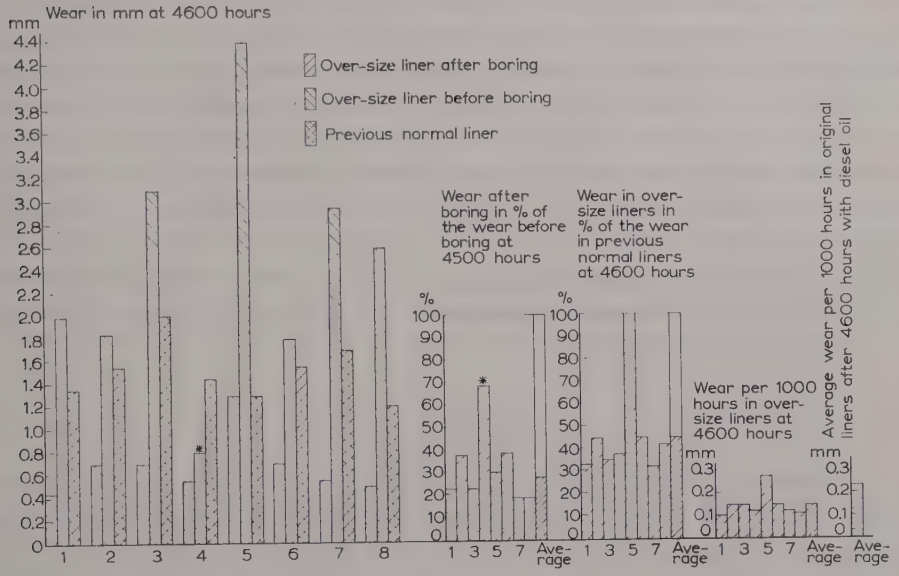
Wear in mm at 4600 hours
mm
Over-size liner after boring
Over-size liner before boring
Previous normal liner
Wear after boring in % of the wear before boring at 4500 hours
Wear in over-size liners in % of the wear in previous normal liners at 4600 hours
Wear per 1000 hours in over-size liners at 4600 hours
Average wear per 1000 hours in original liners after 4600 hours with diesel oil
%
mm
1 2 3 4 5 6 7 8
1 3 5 7 Ave-rage
1 3 5 7 Ave-rage
1 3 5 7 Ave-rage
Ave-rage

Fig. 19. Cylinder wear. M/S *"Paraguay"*. DM 50-VF-90. Used cylinder liners from different ships bored at over-size 506.5 mm and placed in M/S *"Paraguay"* January 1957 as substitute for liners with the normal boring, 500 mm.

* The wear in cylinder 4's liner boring at over-size in connection with diesel oil. The other wear results are in connection with mixed oil (50%).

The 3 curves in Fig. 18 reveal a number of facts. The solid line curve shows how the worn, bored cylinder was wearing before it was bored. The dotted curve shows how the cylinder now bored is wearing in the engine, and finally the dot-dash curve shows how the preceding cylinder was wearing. Quite an essential improvement in the wearing results will be noticed. Operating conditions are the same. The sailing of the vessel is the same. The mixed oil is the same. Rather characteristic is cylinder No. 5, which, before it was bored, was a bad cylinder. After boring it has improved under the working conditions to which it is now subjected, but is still not so good as the others. From this it may be concluded that the wearing conditions in this cylinder must be related in some way to the composition of the material of the liner proper. If the average result of the wear in all the liners is considered, it will be found that, under normal conditions with an ordinary straight mineral cylinder oil, it is possible, using a heavy oil as fuel oil, to reduce the average wear to 0.15 mm per 1000 hours — a result which is fully in agreement with what has been recorded, for example, for the *"Oklahoma"*

and the "*Ohio*", which operate on diesel oil and are lubricated with additivated cylinder oils.

It is unlikely that the reason for this improvement in the wearing properties is that the liners have been used in other engines, and in consequence that structural changes in the material have taken place. Such an explanation is also contradicted by the fact that no similar improvements have been recorded for any other engine that is running similarly with bored liners. The fact that the cylinder diameter has become $6\frac{1}{2}$ mm larger has practically no influence upon the cylinder wear. The most essential cause of the improved result is probably to be sought in the cooled scavenging air. Firstly, a lower combustion temperature is obtained; a drop in the scavenging temperature of 25°C gives a drop in the combustion temperature of almost 75°C. Secondly, improved cooling of the cylinder surfaces is obtained, and with cooler cylinder surfaces a more resistant and stable oil film is obtained, which acts as a protection for the cylinder surface.

One fact, however, is established, *viz.* that in only one case during a period of 6000 hours was it possible with comparatively small changes to reduce the wear from 0.38 to 0.15 mm per 1000 hours.

CONCLUSIONS

To draw any reliable conclusions at this juncture from the punched cards recording of the cylinder wear must be said to be too hazardous. The available observations are not sufficient for this.

Certain assumptions have been strengthened, and the "normal curves" for the individual groups of ships afford a good medium for drawing comparisons between the groups of ships with respect to economical and technical details.

In order to reduce the standard deviation, which makes it difficult to draw conclusions in connection with more detailed tests, *e.g.* the influence of the composition of the materials on the wear, it is intended in future to eliminate the influence of the running-in wear, since this depends to an essential degree on indeterminate and uncontrollable factors.

Therefore, it is further intended in the case of each individual liner to consider only the wearing progress after 3000 hours' operation — *i.e.* to adopt the 3000 hours' wear as zero point, and with this as starting point to work out new normal curves in which the linear regression and not the fluctuating average will be applied as a formula of equalization.

The justification for this decision is:

(1) With the sliding average, 3000 hours are lost, the first point on the curve being at 6000 hours.

(2) If a start is made at the 3000 hours point instead of the original zero point, there will be a higher degree of proportionality between the wear and the number of hours, and consequently the linear regression will be more advantageous as a formula of equalization. The origin of the curve can be at the 3000 hours point

and a definite parameter is obtained, which can be applied as a basis for the comparison of the groups of ships, and also as a regression coefficient which affords a unique accuracy assessment of the parameter.

At present only the elementary stage has been reached and further progress can be made only if engine-manufacturers and oil-suppliers, without regard to competition, will supply the interested parties with details as complete as possible.

If this is done, the author is convinced that in the course of time, when the recorded observations are sufficiently extensive, it will be possible to arrive at a deeper understanding of the conditions of wear and — above all — the degree of dependence on the composition of materials.

The background for this optimism is that punched cards registration furnishes the desired information rapidly and conveniently, and that with the aid of the electronic calculating machine accurate computations can be made that would otherwise be impracticable.

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REVIEW ON THE FACTORS INFLUENCING THE FRICTION OF FIBRES, YARNS AND FABRICS

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SYMBOLS

μ	Coefficient of friction
μ_s	Coefficient of static friction
μ_k	Coefficient of kinetic friction
μ_{RT}	Coefficient of friction in the direction root-to-tip
μ_{TR}	Coefficient of friction in the direction tip-to-root
T_1	On-going tension
T_2	Off-going tension
θ	Angle of lap
ϱ	Radius of curvature of bollard
F	Frictional force
L	Normal load
A	Apparent area of contact
S_0	Shear strength of contact
σ	True compressive stress
ε	Natural compressive strain
b	Stress–strain modulus
x	Stress–strain index, *i.e.* the stress–strain curve of the material can be represented by $\sigma = b\varepsilon^x$

INTRODUCTION

The frictional properties of yarns and fibres have proved important at all stages in the manufacture of textiles. Examples are the importance of fibre friction in wool felting, in carding and drafting during spinning and of yarn friction in the winding, weaving and knitting processes. Friction is also of importance in determining the properties of finished products such as the breaking strength and elastic properties of yarns and fabrics, the "handle", tear strength, abrasion resistance and stability of structure of fabrics. It is for these reasons that investigations of the frictional

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behaviour of textile materials have been undertaken. Few reviews of textile friction have been made to date; BOWDEN AND TABOR[1] devote a section of their book to the friction of fibres, KASWELL[2] devotes a chapter to friction, while a literature survey on fibre friction has been conducted by LANGTON AND RAINEY[3], who reviewed the methods available for measuring the frictional properties of fibres. CORMANN[4] has discussed the factors affecting the friction of textile fibres and cites twenty-four literature references up to 1955. In the present review, it is proposed to discuss the frictional behaviour of fibres, yarns and fabrics and to examine the theories put forward to explain this behaviour.

DEPARTURE FROM AMONTONS'S LAWS

It has been known for a considerable time, that textile materials do not obey Amontons's laws of friction. Thus, MORROW[5], in 1931, found that the force necessary to withdraw a tuft of parallel cotton or viscose rayon fibres placed between two similar tufts depended on the normal pressure and on the apparent area of contact. MORROW also examined the friction of a yarn traversing a cylindrical object transversely and determined the coefficient of friction, μ, using the "bollard" friction equation $T_2/T_1 = e^{\mu\theta}$, where T_1 and T_2 are the on-going and off-going tensions in the yarn and θ is the angle of lap (it should be noted that this equation is valid only for a material which obeys Amontons's laws of friction).

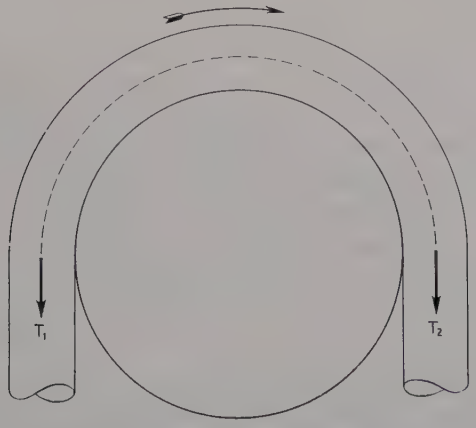

Fig. 1. Diagram illustrating a yarn traversing a cylindrical bollard transversely. When Amontons's laws are obeyed $T_2/T_1 = e^{\mu\theta}$. When $F = CL^{\beta}$, the tensions are related by the equation $T_2^{1-n} - T_1^{1-n} = K(1-n)\theta^{1-n}\theta$.

He observed that the coefficient of friction was dependent on T_1 and pointed out that this indicated an increase in the coefficient of friction with decreasing pressure. Using a similar experimental technique, in which yarns were made to traverse cylindrical bollards transversely, these results were confirmed by BREAZEALE[6] using

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rayon tyre cord running against a glass cylinder, and by BUCKLE AND POLLITT[7] with viscose rayon and cotton yarns lapping a stainless steel cylinder. MARTIN AND MITTELMANN[8] measured the friction of wool and mohair fibres sliding around wool- and mohair-covered cylinders. Here again, the large dependence of the coefficient of friction on T_1 was demonstrated, indicating a departure from Amontons's laws. This was confirmed when the apparent area of contact was altered (i) by using bundles made up of different numbers of fibres, when the coefficient of friction was shown to increase with an increase in the number of fibres in the bundle and (ii) by varying the diameter of the fibres when, for both wool and mohair, the coefficient of friction was shown to increase with increase in the fibre diameter. These animal fibres show a directional frictional effect (D.F.E.) in that the coefficient of friction in the direction tip-to-root, μ_{TR}, is greater than the coefficient of friction in the direction root-to-tip, μ_{RT}. The D.F.E. will be discussed in greater detail in a later section.

Using a "stick-slip" method with two viscose rayon fibres crossed at a small angle, GRALÉN AND OLOFSSON[9] demonstrated the dependence of the static and kinetic coefficients of friction on the normal load. Using a different experimental technique in which two fibres were twisted together and the force necessary to slide the fibres was measured, LINDBERG AND GRALÉN[10] demonstrated the departure from Amontons's laws with nylon and wool fibres. MERCER AND MAKINSON[11] adopted a "stick-slip" technique in which the fibre was drawn over a cylindrical object and here, again, the coefficient of friction was shown to increase as the normal load was reduced, for the following combinations: wool on horn, wool on glass, nylon on nylon, silk on silk and rubber on rubber.

In 1951 HOWELL[12] published the results of his experiments with crossed fibres of viscose, cuprammonium and acetate rayon and nylon; the now familiar dependence of the coefficient of friction on the normal load and on the area of contact being confirmed. Further evidence was furnished by the experiments of WOOD[13], who measured the kinetic friction of viscose rayon fibre using a technique originated by POSTLE, INGHAM AND COX[14], in which a pad of parallel fibres was compressed in a rectangular channel and the force necessary to withdraw one of the fibres was measured. Using an inclined plane apparatus, BANDYOPADHYAY[15] investigated the friction of jute and allied fibres and showed that the coefficient of friction increased as the normal pressure was reduced.

Until this time, workers in the textile friction field[5,9–13,15] had fitted their data to the empirical equation $F = KL + mA$ or its equivalent, $\mu = K + mA/L$. In 1951, BOWDEN AND YOUNG[16] found that the relation between the friction force F and the normal load L, for clean diamond sliding on diamond could be closely represented by the equation $F = CL^{\beta}$, where the index $\beta < 1$, and this equation was subsequently shown by GUTHRIE AND OLIVER[17] to represent their measurements of the fibre-to-fibre friction of viscose rayon very well. LINCOLN[18] showed that this equation was applicable to nylon and, further, that results previously published by HOWELL[12] conformed to this equation. HOWELL AND MAZUR[19], CHAPMAN, PASCOE AND TABOR[20], PASCOE AND TABOR[21], LORD[22] and HUFFINGTON[23] with similar fibres and MAZUR[24]

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with dissimilar fibres have shown that this power law expresses the relation between the frictional force and the normal load very closely.

HOWELL[25] showed that the off-going tension developed in a nylon yarn as it traversed a perspex or glass cylinder was dependent on the radius of curvature of the cylinder. Similarly, BAIRD AND MIESZKIS[26] showed that the coefficient of friction (calculated from $T_2/T_1 = e^{\mu\theta}$) increased as the cylinder diameter increased and as T_1 decreased.

HOWELL[27] and, independently, LINCOLN[28] recalculated the "bollard" friction equation for a material obeying the friction law

$$F = CL^{\beta} \tag{1}$$

and both obtained the relation

$$T_2^{1-\beta} - T_1^{1-\beta} = C(1-\beta)\varrho^{1-\beta}\theta \tag{2}$$

where ϱ is the radius of curvature of the bollard. More recently, LODGE AND HOWELL[29] and RUBENSTEIN[30] have shown theoretically that C and β are dependent on the geometrical configuration of the contacting surfaces. Now, while eqn. (1) is usually obtained experimentally either with a sphere sliding on a flat surface or with two crossed cylindrical fibres, eqn. (2) applies to the case of a yarn lapping a cylindrical object transversely, and the two configurations are geometrically dissimilar. Thus, for generality, eqn. (2) may be written

$$T_2^{1-n} - T_1^{1-n} = K(1-n)\varrho^{1-n}\theta \tag{3}$$

where, in general, $K \neq C$ and $n \neq \beta$.

Thus, the frictional behaviour of textile materials is specified by two parameters, K and n, and only in the case when $n = 1$ (then $K = \mu$) does the coefficient of friction have any physical meaning. It is possible, however[31,32], to define an "effective" coefficient of friction, μ_e, (which depends on the initial tension, T_1, and on the bollard radius, ϱ) and then formulae derived for cases where Amontons's laws hold can be applied in cases where Amontons's laws are not obeyed. In the papers cited above, it is shown that $\mu_e = (\alpha/\psi)\ \log_e\ (1+\psi)$, where $\alpha = K(\varrho/T_1)^{1-n}$ and $\psi = (1-n)\ \alpha\theta$. When ψ is sufficiently small, $\mu_e \triangleq \alpha$. Now ψ is small when $n \triangleq 1$ and/or when θ is small. The case when $n \triangleq 1$ was considered by HOWELL[27], while RUBENSTEIN[31] has considered both cases and has shown that the equation $T_2/T_1 = e^{\alpha\theta}$ can be applied without introducing errors in T_2/T_1 greater than 5% provided

$$\alpha\theta \leqslant 0.5.$$

Equation (3) has been verified experimentally in several ways in recent years. Thus T_2 has been measured (i) with ϱ,θ constant and T_1 varied; with wool fibre[28], with cellulose acetate yarn[33] and with nylon, viscose, cellulose acetate and cotton yarns[34] (ii) with ϱ,T_1 constant and θ varied; with cotton, nylon and cellulose acetate yarns[34] and (iii) with θ,T_1 constant and ϱ varied; with cellulose acetate yarn[33] and

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with nylon and viscose rayon yarns[31]. Finally, MACK[35] has considered the case of a string traversing a cylindrical object slantwise and the resulting relation between T_1 and T_2 has been verified experimentally by MACK AND RUBENSTEIN[32].

Thus, a considerable body of experimental evidence has accumulated demonstrating that Amontons's laws are not obeyed by yarns and fibres and, empirically, it has been found that the power law relation between the frictional force and the normal load has proved most successful. As will be seen later, this power law has been derived, with the aid of simple assumptions, as a consequence of the adhesion theory of friction and of the mode of deformation of these materials.

DEPENDENCE OF THE FRICTIONAL FORCE ON FIBRE DIAMETER

It follows from the fact that the frictional force is dependent on the apparent area of contact that the frictional force should depend on fibre diameter. It has been mentioned above that MARTIN AND MITTELMANN[8] noted an increase in the coefficient of friction of wool and mohair with increase in fibre diameter. Similarly, GRALÉN[36] quoted results obtained with Terylene fibres which suggested an increase in the coefficient of friction with increase in fibre diameter. In view of the small variation in fibre diameter ($13.5 - 18.3\ \mu$), GRALÉN felt that this effect required further experimental verification. This has been furnished by the experiments of PASCOE AND TABOR[21] who measured the friction of crossed nylon fibres of three different diameters and the friction between a nylon hemisphere and a flat nylon surface. The specimen diameter was varied in this way over a range of 320:1 and in this range the frictional force was found to increase with increase in specimen diameter.

DEPENDENCE OF THE FRICTIONAL FORCE ON SURFACE ROUGHNESS

KING[37] showed that scraping, and thus roughening, a cylindrical surface about which a wool fibre was drawn produced a reduction in μ_{RT}, although μ_{TR} showed no reduction. LINCOLN[18] repeated these experiments using nylon filaments and stainless steel surface with different degrees of roughness and again showed that the coefficient of friction decreased as the roughness increased. BAIRD AND MIESZKIS[26] investigated the influence of bollard roughness on the coefficient of friction of a nylon yarn. They used alumina cylinders of five different degrees of roughness and showed that as the roughness increased, the coefficient of friction decreased in general but that with the roughest object, there was a tendency for the coefficient of friction to increase, which latter effect they attributed to the interlocking of asperities on yarn and friction object, *i.e.* to a Coulomb frictional component. The decrease in frictional resistance with increase in roughness was confirmed when they varied the twist inserted in a multifilament nylon yarn and showed that as the twist, and hence the surface roughness, of the yarn was increased, the coefficient of friction decreased — an effect also noted by LYNE[33] with cellulose acetate yarn. Finally, BAIRD AND MIESZKIS investigated the effect of the titanium dioxide content* of a nylon yarn on the coeffi-

* Titanium dioxide is normally incorporated as a delustrant.

cient of friction and found that as the titania content and, presumably, the surface roughness increased, the coefficient of friction decreased.

When eqn. (3) was derived from first principles, RUBENSTEIN[34] showed that if the surface roughness of a cylindrical object were increased, K would decrease while n would probably increase but could remain unchanged. Experimentally, these conclusions were confirmed with nylon, viscose rayon and cotton yarns traversing both smooth and ground glass rods, when it was found that in all cases n increased and K decreased when the smooth object was replaced by a rough one. When two viscose rayon yarns, nominally identical in all respects save that one was bright while the other was delustred with titanium dioxide, were compared using a smooth glass object, it was found that the values of n were the same and that the change in K was within the limits of experimental error. These results, in contrast to those of BAIRD AND MIESZKIS mentioned above, confirmed the results of GUTHRIE AND OLIVER[17], who compared the fibre-to-fibre friction of viscose rayon, and it was suggested[34] that this difference resided in the fact that the nylon filament is smooth and the slight roughening effect produced by an increase in the titania content is detectable but that the filament of viscose, being crenulated, is, in all probability, too rough for the slight increase in surface roughness produced by delustring to cause a detectable change in the frictional resistance.

This decrease in the frictional force as surface roughness increases has been shown theoretically[30] to occur when Amontons's laws are not obeyed, the frictional force being independent of roughness when Amontons's laws are obeyed; this latter conclusion was verified experimentally for metals by STRANG AND LEWIS[38] and has been noted by BOWDEN AND TABOR[1] (pp. 78, 175). It is interesting, at this point, to note that these conclusions were derived assuming the validity of the adhesion theory of friction, while a corollary to the Coulomb theory of friction is that the frictional force increases as the surface roughness increases[39]. Thus, the experimental evidence of the effect of roughness, obtained with yarns and fibres, is seen to be in accordance with predictions based on the adhesion theory and contrary to the predictions of the Coulomb theory of friction.

DEPENDENCE ON HUMIDITY AND MOISTURE CONTENT

MORROW[5] investigated the effect of the moisture content of a cotton yarn on the coefficient of friction as the yarn traversed a steel rod and showed that as the moisture content increased, the coefficient of friction increased considerably. These results were confirmed by MORROW when the relative humidity of the atmosphere surrounding the apparatus was varied between 0% and 100% and the coefficient of friction increased from 0.27 to 0.44. POLLITT[40] discussed the importance of humidity on certain yarn properties which influence knittability. Among these factors, the influence of humidity on the coefficient of friction of yarns was considered and the importance of this effect, particularly at relative humidities about 80%, was emphasised. FRISHMAN *et al.*[41]. measured the coefficient of friction of wool fibres both wet

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and dry and found the coefficient of friction determined in the wet state to be higher than that determined with dry fibres. BANDYOPADHYAY[15] investigated the influence of relative humidity on the coefficient of friction of jute fibres and showed that in the range 40%–80% r.h., the coefficient of friction increased as the relative humidity increased. WOOD[42] found only a small variation with viscose rayon fibres in that, as the relative humidity was increased from 30% to 80% the frictional force increased by 15%. KING[37] and later LINCOLN[18] noted that the elastic properties of fibres vary with relative humidity as a result of moisture sorption and they suggested that the frictional properties should exhibit a similar variation. KING noted an increase in the friction of both wool and nylon fibres with increase in moisture content and LINCOLN confirmed this increase with nylon.

The large increase in the frictional force with increase in moisture content of cotton at room temperature noted by MORROW was not confirmed when a cotton yarn traversed a chromium-plated metal cylinder heated from 20°C to 140°C[34]. Two experiments were performed, the first at constant relative humidity, so that as the bollard was heated to higher temperatures the yarn became drier, and the second at constant average moisture content, which was achieved by progressively raising the relative humidity as the bollard temperature was raised. It was found that the value of n remained constant for both experiments, while there was very little difference in the variation of K with the temperature determined under the two conditions.

THE INFLUENCE OF TEMPERATURE ON FRICTION

MORROW[5] determined the influence of temperature on the coefficient of friction of a cotton yarn when the yarn was in a dry atmosphere and when the yarn was in a moist atmosphere. For both conditions, he noted a decrease in the coefficient of friction with increase in temperature but he suspected that part, at least, of the reduction occurring with the moist yarn was attributable to the drying out of the yarn as the temperature of the atmosphere was raised. LINDBERG AND GRALÉN[10] showed that the friction of wool fibres depends on temperature, both μ_{TR} and μ_{RT} decreasing as the temperature is raised.

The influence of temperature on the parameters K and n in eqn. (3) was investigated by RUBENSTEIN[34] with nylon yarn, cotton yarn which contained natural fats and waxes (*i.e.* grey cotton) and cotton yarn from which the natural fats and waxes had been removed by scouring. In this work, n was shown to depend on the stress–strain index of the material, being a minimum for an elastic material and a maximum for a plastic material. The relation between n and the stress–strain index, x, was shown to depend on whether the yarn–bollard contact could be regarded as smooth or rough and, when rough, on the shape of the asperities. With the assumptions that when the contact was rough (i) the asperities were all of the same shape and all protruded from the surface by the same amount, and (ii) the stress–strain index of the material of the yarn asperities, x_1, is proportional to that of the bulk material of the yarn, x, it was shown that for a given value of x_1, $n_{rs} > n_{rc} > n_s$ where

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n_{rs} is the value of n when the asperities are of spherical shape
n_{rc} is the value of n when the asperities are cylindrical and
n_s is the value of n when the contact is smooth.

For a perfectly plastic material $n_{rs} = n_{rc} = n_s = 1$, while for a perfectly elastic material, provided $x_1 = x\ (= 1)$,

$$n_s = {}^1/_2, \quad n_{rc} = {}^3/_4 \quad \text{and} \quad n_{rs} = {}^5/_6.$$

For nylon, a thermoplastic material, it would be expected that as the temperature was increased, n would increase and this was verified experimentally. Cotton, however, is not thermoplastic and a much smaller variation in n would be expected — in fact, within the limits of experimental error, n was shown to be constant over the temperature range 20–140°C.

The value of K was shown to depend on the ratio S_0/b where S_0 is the shear strength of the contact and b is the stress–strain modulus. For a lubricated contact, S_0 is determined largely by the shear properties of the lubricant and b is determined by the deformational properties of the yarn, and thus K will decrease, remain constant or increase with increase in bollard temperature according as S_0 decreases faster than, at the same rate as, or slower than b decreases with increase in temperature. With lubricated nylon yarn, K decreased rapidly between 20°C and 70°C, remained steady between 70°C and 80°C and increased slowly between 80°C and 140°C. Similarly, with grey cotton yarn K decreased rapidly between 20°C and 120°C and increased only slightly between 120°C and 140°C. With wax-free cotton yarn, where both S_0 and b are determined by the properties of cotton, the ratio S_0/b can be expected to vary only slowly with temperature since, as already indicated, the mechanical properties of cotton are not greatly affected by temperature. Experimentally, over the range 20–145°C the value of K was found to lie in the range 0.78 ± 0.02, in contrast to the two-fold variation in K noted with grey cotton.

THE EFFECT OF VELOCITY ON FRICTION

The coefficient of friction of cotton yarn has been measured[3] in the velocity range 0–600 m/min and over this range the coefficient of friction was found to increase with increase in velocity. BUCKLE AND POLLITT[7] obtained similar results with cotton, cellulose acetate, silk and viscose rayon yarns in the range 0–130 m/min. Over a smaller range, *viz.* 6–27 m/min, WOOD[13] showed that the frictional force required to withdraw a viscose rayon filament from a bundle of parallel fibres increased with increase in velocity. GRALÉN AND OLOFSSON[9] working at very low velocities (< 1 m/min) with single fibres were unable to detect any increase in the coefficient of friction with increase in velocity. BAIRD AND MIESZKIS[26] determined the coefficient of friction of nylon yarn in the velocity range 90–450 m/min. Against chromium-plated steel and against a glass rod the coefficient of friction was shown to increase, but against a ceramic material the coefficient of friction was approximately constant as the velocity was increased.

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LYNE[33], using lubricated cellulose acetate yarn measured the off-going tension as the yarn traversed a chromium-plated cylinder and showed that the tension increased rapidly up to a speed of 150 m/min and then less rapidly from 150–400 m/min. In the same work, LYNE demonstrated the increase in T_2 with increase in the viscosity of the lubricant. These results have been analysed by HANSEN AND TABOR[43] who have shown that the dependence of T_2 on lubricant viscosity and on velocity is consistent with the lubrication in this case having been largely hydrodynamic. This has led them to suggest that previously quoted cases of the coefficient of friction of textile materials increasing with increase in velocity may have been due to hydrodynamic lubrication and this could well be true in some instances. However, when the off-going tensions developed in four yarns which had been freed from contamination by lengthy extraction with organic solvents and with water were measured[34] in the range 0.0001–58 m/min it was found that at very low velocities (0.0001–0.001 m/min) T_2 decreased with increase in velocity (this was confirmed by the evidence of a "stick-slip" motion at these velocities). Above 0.001 m/min T_2 increased with increase in velocity; thus, in these particular cases where the yarns were virtually lubricant-free, there can be no question of the increase in friction with increase in velocity being due to hydrodynamic lubrication.

When the values of K and n for cotton and nylon yarns were determined[34] in the range 0–66 m/min it was found that n decreased while K increased with increase in velocity. An explanation of these observations was offered in terms of the visco-elastic properties of these materials.

THE DIRECTIONAL FRICTIONAL EFFECT (D.F.E.)

It has been noted above that most fibres of animal origin exhibit a D.F.E. in that μ_{TR} is greater than μ_{RT}, a fact first reported by MONGE[44] who attributed it to the overlapping scale structure of the surface of the fibre. This gave rise to the "ratchet theory" which was used by MERCER AND MAKINSON[11], MAKINSON [45] and LINDBERG AND GRALÉN[10] to explain the D.F.E. According to this theory, motion against the scales results in the scale edges interlocking, so that to produce motion in this direction it is necessary to deform or rupture the interlocked region.

Several investigations[46–48] have shown that anti-felting agents (*e.g.* chlorine, sulphuryl chloride) reduce the D.F.E. of wool fibres, and this led MARTIN[48] to attribute the D.F.E. to an asymmetrical field of force at the surface of the fibre. LIPSON AND HOWARD[49], who rubbed wool fibres on horn treated with reagents used commercially for shrink-proofing, found that these treatments reduced the D.F.E. In view of the fact that it was not necessary to treat the fibres themselves in order to reduce their D.F.E., LIPSON AND HOWARD felt that their results supported the view that interlocking of the scales was not the main factor responsible for the D.F.E.

THOMSON AND SPEAKMAN[50] found, on coating wool fibres with thin films of silver or gold, that the D.F.E., though reduced, still persisted and photo-micrograms showed that the scale structure was still visible. These results, together with the

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fact that the D.F.E. is evident with lubricated fibres (*e.g.* BOHM[51]) are opposed to the explanation put forward by MARTIN.

More recently, two independent and complementary explanations of the D.F.E. of wool fibres have been proposed. The first explanation, offered by LINCOLN[28] to account for the D.F.E. of dry fibres, assumes an idealised scale shape and assumes that the true areas of contact are produced by elastic deformation. GROSBERG[52] assumes the wool fibre to be anisotropically elastic, deforming more readily at an angle to the scale than at right angles. With these assumptions, GROSBERG accounts for the large D.F.E. associated with wet, swollen fibres and shows, further, that while the ratchet effect could produce an important contribition to the D.F.E. of dry fibres, the contribution in the case of wet fibres, where the D.F.E. is large, would be negligible.

THE SURFACE FRICTION OF FABRICS

RENNIE[53], in 1829, investigated the frictional behaviour of several materials, including cloth, by applying a tangential force to a loaded block and by an inclined plane method. As a result of experiments with this latter method, RENNIE found that the angle of repose increased when the normal load was reduced and when the apparent area of contact was increased. These, however, are not the earliest recorded friction experiments with cloth, for RENNIE makes a reference to work done by VINCE in 1784.

MERCIER[54] described a simple method of specifying the slipperiness of fabrics by measurement of the coefficient of friction between two pieces of the same material with an inclined plane apparatus: the slider consisted of a block covered with a sample of the fabric, while the inclined plane was covered with another piece of the fabric. It was found that during motion, rearrangement of the surface fibres of the fabric occurred so that successive determinations resulted in a decrease in the angle of repose to a constant limiting value, the tangent of which was taken as the coefficient of friction. MORROW[5] developed a method for the determination of the coefficient of kinetic friction, in which a strip of fabric was fastened to the rim of a wheel run at constant speed and another piece of cloth was mounted on a block which rested on the rim under a known load. From observations of cloth sliding on a similar cloth, MORROW found a similar dependence of the friction force on load and on apparent area as he had noted with fibres.

DREBY[55] produced an instrument for measuring the coefficient of kinetic friction of fabrics. He found that the coefficient of friction decreased as the distance of slip increased. This decrease, which DREBY attributed to compacting and surface-smoothing effects, continued with successive remeasurements of the coefficient of friction. It was also noted that an increase in velocity produced an increase in the coefficient of friction. Recently, LAVRAKAS[56] has determined the frictional properties of parachute cloths at high sliding velocities using a belt-friction apparatus. In this work it was noted that the coefficient of kinetic friction decreased as the load increased and as the relative humidity decreased.

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MIURA[57] has investigated the friction of cotton fabrics and has shown that the frictional force per unit apparent area is related to the normal pressure by a power law of the same form as has been found to apply to fibres and yarns. Thus, while the phenomena associated with the surface friction of fabrics are complicated by changes in the materials during sliding, it would appear, as might be expected, that the frictional characteristics of fabrics closely resemble those of fibres and yarns.

THE EFFECT OF LUBRICATION

With carefully cleaned yarns in contact with smooth, clean surfaces, very high resistance to sliding can be encountered[58]. The application of lubricants to yarns and fibres reduces the coefficient of friction to a value between 0.35, in the case of a poor lubricant, and 0.10 in the case of a good lubricant. Despite the fact that technological problems arise, in general, from frictional forces which are too high rather than from frictional forces which are too low, the lubrication of textile materials has not been widely investigated.

MOSS[59] determined the lubricating effect of a large number of substances on cotton yarn. In general, it was found that as the amount of lubricant applied to the yarn was increased, the coefficient of friction decreased rapidly from 0.65 (the unlubricated value) to a value between 0.15 and 0.35, depending on the lubricant applied. This decrease was normally completed by the time 1.5% of lubricant (by weight based on the weight of unlubricated yarn) had been applied to the yarn: often 0.5% or less was sufficient, although in one case (castor oil soap) 3.0% proved necessary. After this initial decrease, further addition of lubricant (up to 4%) could produce either no further change in the coefficient of friction (*e.g.* methyl stearate), a slow increase in the coefficient of friction (*e.g.* paraffin wax) or a rapid increase in friction (*e.g.* octadecyl alcohol). Of the substances tested, paraffin wax, methyl stearate and methyl oleate proved the most effective lubricants for cotton.

RÖDER[60] investigated the effect of a large variety of lubricants and commercial finishing agents on the frictional properties of viscose rayon fibre. By measuring the static (μ_s) and the kinetic (μ_k)* coefficients of friction, RÖDER was able to classify the finishing agents into three groups according to the value of $\mu_s - \mu_k$ they produced when applied to the fibre. Each group was found to contain agents of a similar chemical nature, *i.e.* group I included non-ionic agents, group II, anionic-active agents and group III, cationic-active agents. The effect of speed was quite different when the fibre was "dressed" with agents from different groups in that, with no finish or with finishing agents of group I, an increase in speed produced an initial decrease in μ_k followed (at velocities higher than ~ 2 m/min) by a monotonic increase in the coefficient of friction, while with agents from groups II and III, the coefficient of friction increased monotonically with increase in velocity. Similarly, the effects of humidity and of amount of finish applied to the fibre were shown to depend on the group to which the finishing agent belonged. The "handle" of fibres was shown

* The speed at which μ_k was determined was arbitrarily chosen at 0.9 m/min.

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by RÖDER to depend on the difference between μ_s and μ_k, the fibres (i) being extremely slippery if $\mu_s - \mu_k$ were negative, (ii) being normally soft if $\mu_s - \mu_k$ were positive but less than 0.015 and, (iii) having a harsh handle if $\mu_s - \mu_k$ were greater than 0.015. In a later paper[61] these ideas were developed to correlate the frictional properties of yarns and fibres with their behaviour during processing.

The results obtained by LYNE[33] with lubricated cellulose acetate yarn and the interpretation, in terms of the establishment of hydrodynamic lubrication, by HANSEN AND TABOR[43] have been referred to above. In addition to the viscous drag set up by this mechanism, the surface tension of the lubricant can produce an adhesion force between the fibre or yarn and the bollard which, if ignored, can produce seriously erroneous values of the coefficient of friction[62]. Both of these effects decrease as the radius of curvature of the bollard decreases and as the roughness of the contact increases. For this latter reason, RUBENSTEIN[34] used only spun yarns to determine the influence of lubrication on the values of the parameters K and n. The value of n was found to be independent both of the lubricant used and of the amount of lubricant applied to nylon yarn. For nylon and cotton yarns lubricated with methyl stearate and with a commercial mineral oil, K was shown to decrease with increase in the amount of lubricant added to the yarn until a minimum value was reached, after which further addition of lubricant produced no change in the value of K. These results were shown to be consistent with the effect which the application of a lubricant could be expected to produce on the stress–strain index, x, the stress–strain modulus, b, and the shear strength of the contact region, S_0.

Currently an investigation is in progress at these laboratories, to determine the effects produced by the lubrication of yarns and to obtain some information on those properties of a material which make it a good lubricant for textile purposes.

THEORETICAL INTERPRETATIONS OF TEXTILE FRICTION

Up to 1952, workers in the field of fibre friction had fitted their frictional force–normal load data to linear relations of the form $F = KL + mA$ or the equivalent $\mu = K + mA/L$. The additional term, mA, introduced to account for the departure from Amontons's laws, was attributed by MORROW[5] to adhesion produced either by sticking of the waxy surfaces or, more probably, by the interlocking of the convolutions of the cotton fibre. GRALÉN AND OLOFSSON[9] and OLOFSSON[63] attributed the additional term to an adhesive force but did not specify any particular mechanism to explain it, while HOWELL[12] considered possible mechanisms by which this additional term could be explained.

GRALÉN, OLOFSSON AND LINDBERG[64] derived a linear relation between the coefficient of friction and the reciprocal of the normal load by assuming the material to have a well-defined yield stress and the specimens to be smooth. With this model, the area of contact consists of a central zone produced by plastic deformation and an outer zone produced by elastic deformation. In the past, static-loading experiments in compression[18–21] have revealed that over a considerable load range cellulose

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acetate, viscose rayon and nylon obey a single deformation law with no evidence of a yield stress, well-defined or otherwise. Recently, however, LAWS[65] has determined, by interferometric methods, the true area of contact of a nylon bristle loaded against optically flat glass plate and has shown that A is a proportional to L^{γ} where γ increases from 0.78 at low loads to 1 at loads above 100 g. This increase in γ is consistent with an increasing proportion of the asperities being strained beyond the elastic limit until, eventually, plastic deformation predominates.

KING[37] suggested that there was some relation between the deformational properties of rubbing materials and their frictional properties. Taking this a stage further, LINCOLN[18] calculated a power law relation between the frictional force and the normal load for smooth specimens of a Hookean material. Here the index $\beta = 2/3$ and for the geometrical configuration chosen (either a spherical specimen sliding on a flat surface or of two crossed cylinders) this represents the minimum possible value of β (although, as mentioned above, with the geometrical arrangement occurring with a smooth yarn lapping a smooth cylindrical bollard, the lowest possible value of n is $1/2$). LINCOLN showed that his theoretical deductions were in agreement with experimental results obtained with nylon. Later, LINCOLN[66] and ARCHARD[67] considered the influence of multiple contacts at numerous asperities within the apparent area of contact and it was shown, by considering different models with different asperity shapes, that the value of the index β in the power law relation could approach unity while the deformation was still Hookean. LODGE AND HOWELL[29] developed these ideas for an elastic solid and generalised their analysis by dispensing with the assumption that the material was Hookean. PASCOE AND TABOR[21] determined the deformation law of nylon empirically, from static loading experiments, and by assuming that a single contact occurred (*i.e.* that the specimens were perfectly smooth) they were able to derive a relation between the coefficient of friction, the normal load and the diameter of the specimens which fitted their experimental data quite well. RUBENSTEIN[30] derived a solution which was designed to embrace the frictional behaviour of solids in general by assuming (i) a deformation law of the form $\sigma = b\varepsilon^{r}$, (ii) an idealised roughness model, which was subsequently shown by WILLIAMS[68] to be unnecessarily restrictive, and (iii) that the adhesion theory of friction was valid. With these assumptions it was possible to derive (i) a relation between the apparent area of contact, the normal load and the specimen diameter, which was in agreement with experimental data[18–21], (ii) the power law relation between frictional force and normal load which had been used by previous investigators[16–24], (iii) the dependence of the frictional force on roughness and on apparent area of contact. Later[34] the theory was applied to the case of a yarn lapping a cylinder transversely when the relation between the on-going and off-going tensions was derived and the factors affecting the parameters K and n (in eqn. (3)) were investigated.

CONCLUSION

In conclusion it is interesting to note that despite the fact that most of the research performed in the field of friction has been devoted to an explanation of the phenomena

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observed with metals, it is as a result of the investigations made with non-metals and, in particular, with textile materials, that a deeper appreciation of friction phenomena has been acquired. Amontons's law relating the frictional force to the normal load is now seen to be a particular case of a more general friction law. Further, it is now clear that only when Amontons's law is obeyed is the frictional force independent of apparent area of contact and of the number of points of true contact within the apparent area, that is, independent of surface roughness (provided this is not excessive) and of specimen shape and size.

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Brief Notes

SOME ELECTRICAL PHENOMENA IN METAL CUTTING

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The phenomenon of the generation of a thermo-e.m.f. at the seat of cutting has been used for the determination of cutting temperatures since the 1920's (*e.g.* ref.[1-6]), but only recently has its possible bearing upon tool-wear been suggested[7-9]. In addition to any effect of the current on oxidation-rate and thus wear-rate[7], a connection with "exo"-electrons[10] seems probable[7,11]. Again, experiments with an additional current introduced into the closed system tool–workpiece–machine have shown definite influence on tool-wear[12,13]. This might be associated with variations of the thermo-mechanical balance in the contact area, similar to the effects in "electro-mechanical burnishing" of steel[15]. These changes, under favourable conditions, could lead to a relative increase of wear resistance.

In this note the results of a systematic investigation of the dependence of the thermo-e.m.f. magnitude on cutting conditions in fine turning are shown, and some effects of naturally generated and/or additional injected electrical currents are discussed.

The experimental technique was the same as previously used by the authors[9], and the thermo-e.m.f. e plotted against cutting speed v for brass 60/40 when fine-turned

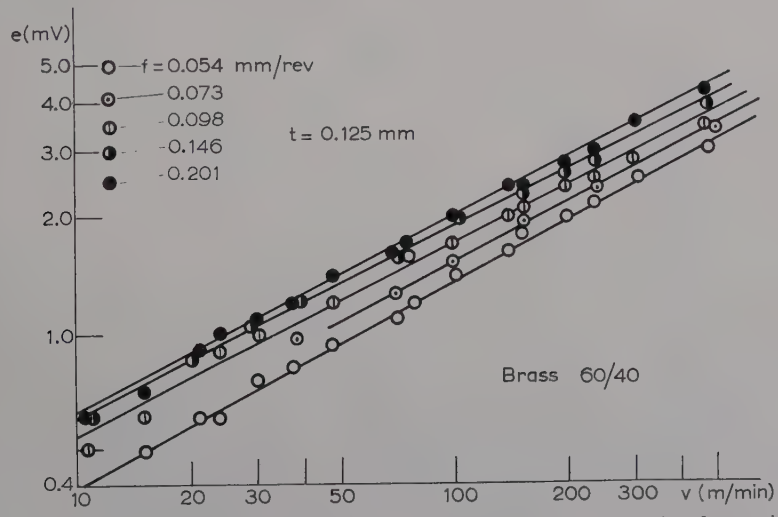

Fig. 1. Thermo-e.m.f. versus cutting speed in finish cutting of brass 60/40 for various feeds.

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with a Mitia TE carbide tool is shown in Figs. 1 and 2 for various feeds f and depths of cut t. The scatter of the individual results is within $\pm 5\%$. They can be approximated by straight lines and expressed by the equation

$$e = av^x, \tag{1}$$

where a and x depend upon cutting factors.

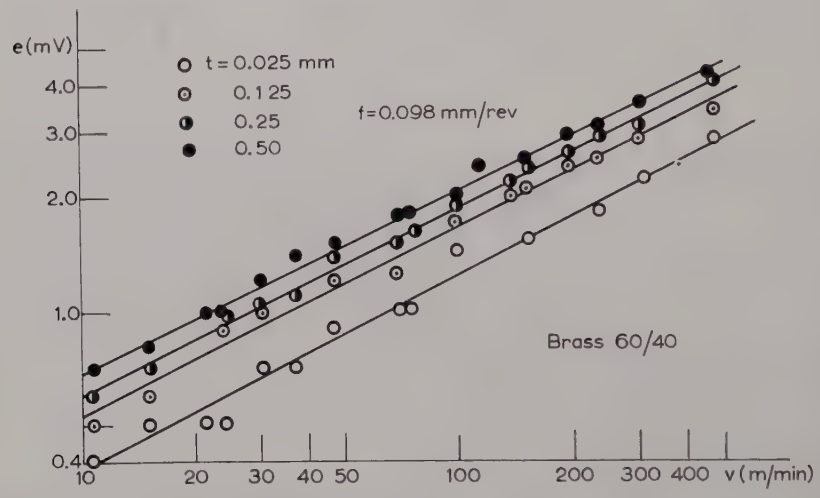

Fig. 2. Thermo-e.m.f. versus cutting speed in finish cutting of brass 60/40 for various depths of cut.

The tests were repeated for 99.8% aluminium, and 0.38% carbon steel and an En 31 alloyed steel, and similar trends were observed. In Figs. 3 and 4 the dependence of the coefficients a and exponents x from eqn. (1) on f and t are shown for the materials

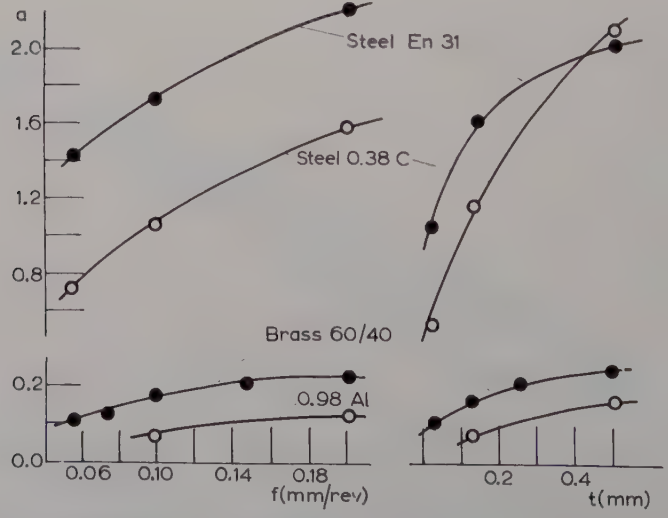

Fig. 3. The coefficient a versus feed, *viz.* depth of cut for materials tested.

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tested. The interesting conclusion is that, in addition to an increase of a with improved mechanical characteristics of the workpiece material; (the temperature increases also), x systematically drops in magnitude, and as a consequence the relative influence of cutting conditions decreases (the mean value for 0.98 aluminium is 0.54; brass 60/40: 0.50; 0.48% carbon steel: 0.38; and for steel En 31: 0.36). In addition, with an increase of chip cross-section, x decreases, and it can be expected that at a sufficiently high speed (by extrapolation one deduces the order of magnitude to be $v \approx$ 1500 m/min), the thermo-e.m.f. for a material should depend on the remaining cutting conditions in a restricted manner only.

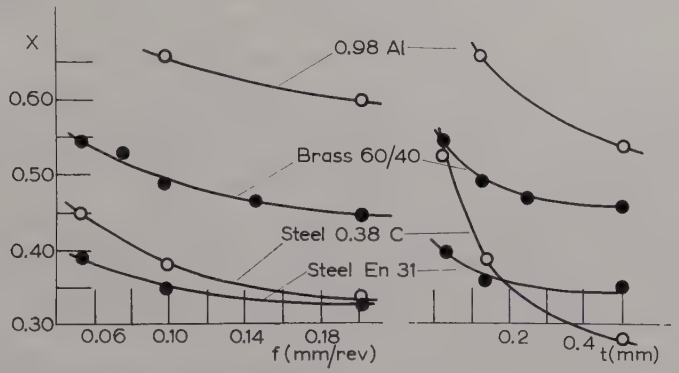

Fig. 4. The exponent x versus feed, *viz.* depth of cut for materials tested.

According to a theoretical analysis[6], the temperature of cutting increases with wear, and this finding has already been used in a method of tool-life control[14]. If a direct relationship existed between variation of the thermo-e.m.f. and slight increases of tool wear, then the more complicated methods employed in tool-wear investigations (*e.g.* the use of a tool microscope or similar device) could be replaced by this more simple method. Unfortunately, a few prolonged tests, similar to those described earlier[16], in which the thermo-e.m.f. was recorded, have shown that in finish turning 0.38% carbon steel ($v \approx 240$ m/min, $f = 0.098$ mm/rev., $t = 0.125$ mm) with a Mitia TE carbide-tipped tool for a total length of cutting $L \approx 2{,}500$ m the wear magnitudes reached $B_1 = 0.38$ mm and $B = 0.20$ mm, whereas the thermo-e.m.f. varied unsystematically within $e = 12.3$–13.8 mV throughout the experiment. In consequence, it does not appear that thermo-e.m.f. measurement provides a reliable method for wear studies in conditions of relatively mild tool-wear.

The electrical phenomena may affect tool-wear in two different ways: (i) the adverse influence of the electrical current caused by the thermo-e.m.f., as found at the Technische Hochschule, Aachen[7], which, however, has not been confirmed at the Technische Hogeschool, Delft[8]; (ii) the favourable influence of additional current (D.C. or A.C.) of high values (400–800 A) and low voltage (0.5–6 V) under severe cutting conditions (*e.g.* form turning of alloyed steel hardened to 62 R_c[12]), owing to the additional heating of the interface.

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Recent work by the present authors[9] appears to provide some additional explanation of the above findings. Under the specific conditions of fine-finish turning of mild steel, no influence of "compensating" current has been observed, thus confirming the conclusions from the Netherlands. However, for ± 40 A (which is, taking into account the ratios of chip cross-sections, of a similar order of magnitude as in Russian experiments[12]) a decrease has been observed in the length of the grooves which form under the trailing edge. This has, as shown recently[16], a direct bearing upon the improvement of surface finish. Although no change of the width of wear-land occurred, the decrease of the intensity of concentrated wear may be explained by the altered effect of the interrelated factors acting at the localized spots. A more detailed experimental study of tool-wear when a current is present, connected with a theoretical study of heat distribution and of electro-chemical effects, might show that throughout the wide range of possible cutting conditions, regimes of cooling or heating of a tool interface exert a changing influence upon tool-life and quality of production. Thus, a widening of the field of the factors of influence in achieving the optimum in machining might occur.

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INFLUENCE OF LOW TEMPERATURES ON THE LUBRICATING CAPACITY OF MINERAL OILS*

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The lubrication properties of lubricant layers on metals are known to be greatly dependent on temperature. This is confirmed by testing the friction of metal surfaces lubricated with oil heated to various temperatures[1,2] and by physical investigations; for example, by studying the structure of hydrocarbon films by the electron diffraction method[3]. When a layer of oil on a metal is heated, it will maintain its lubricity until a critical temperature is reached which varies with the kind of oil and the kind of metal.

The influence of low temperatures on the lubricity of oils has been studied in the wear resistance laboratory of this Institute. The tests were run on a four-ball machine[4] which was modified to produce low temperatures in the oil bath. The plan of the testing machine is shown in Fig. 1. Annealed balls of 12.7 mm diameter and $H_v = 200$ (made of steel, which has 1% C, 1.5% Cr) were used.

The tests were conducted under constant load, the initial contact pressure of surfaces being $q = 170$ kg/mm^2, sliding speed —0.4 mm/sec. The cooling rate of the balls and the surrounding oil was regulated by addition of solid CO_2 to the cooling medium (ethyleneglycol) and registered by a thermocouple and galvanometer "φ".

Each test, carried out at the respective temperature, lasted 60 sec, and in its course the coefficient of friction and the character of sliding (whether smooth or not) were determined.

Two mineral oils differing greatly in viscosity were tested: No. 1, special vaseline oil (MVP), and No. 2, machine oil (MS). The viscosity curves of these oils are shown in Fig. 2.

The results of the tests are presented in Fig. 3, in which the coefficient of friction is plotted against temperature (left-hand side of the diagram). The value of the coefficient of friction is seen to decrease gradually as the temperature is reduced from 20° to —60° for oil No. 1 and to —40° for oil No. 2, respectively. In these experiments wear of the balls, measured after the testing of each oil, was very small.

These results show that: (1) the viscosity of the oil has no influence on the coefficient of friction under the conditions of the experiment; (2) the lubricating capacity (oiliness) of the oil improves with decrease in temperature.

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* Translated by the author from *Doklady Acad. Nauk S.S.S.R., 106* (1956) 494.

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According to the test data obtained when these oils were heated by an external source of heat (right-hand side of the diagram, Fig. 3), the coefficient of friction rises gradually with temperature, increasing to 0.12 for oil No. 1 (at 70°) and to 0.1 for oil No. 2 (at 150°). In the tested temperature range sliding was smooth. At the critical temperature (80° for oil No. 1, and 160° for oil No. 2) and higher the smooth course is interrupted and the coefficient of friction grows by leaps and bounds. In these conditions the wear of balls increases.

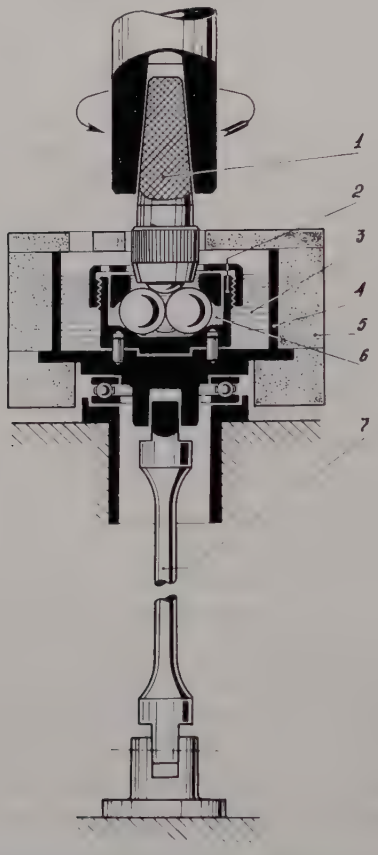

Fig. 1. Plan of four-ball machine for oil testing at low temperatures.
1 – cone made of plastic material
2 – bath containing oil to be tested
3 – ethyleneglycol mixed with solid CO_2
4 – a cap for cooling liquid
5 – insulator
6 – oil to be tested
7 – torsion bar

The data referred to permit us to conclude that mineral oils (free from chemically active additives) act as lubricants (*i.e.* lower the coefficient of friction and reduce the wear of metal surfaces) within a temperature range extending from the freezing point

References p. 317

to critical temperatures, *i.e.* to temperatures at which the bonds formed between the boundary layers of the oils and the metal surfaces are destroyed.

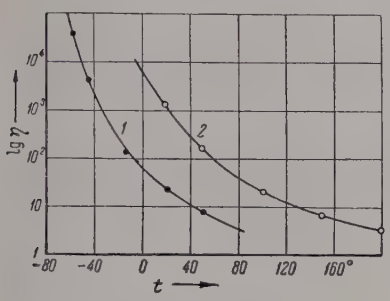

Fig. 2. Effect of temperature on viscosity (Centistokes).
1 – oil No. 1 ("MVP" – in Russian)
2 – oil No. 2 ("MS" – in Russian)

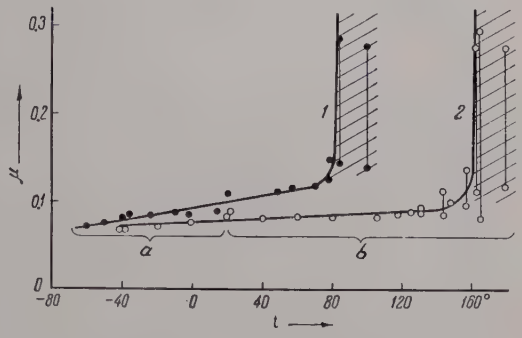

Fig. 3. Frictional behaviour of two different lubricants, as a function of temperatures.
1 – oil No. 1 ("MVP").
2 – oil No. 2 ("MS").
(a) experiment with cooling;
(b) experiment with heating.

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Received November 25, 1958

Recent Russian Literature

A full translation of each of the following papers can be obtained from Henry Brutcher, Box 157, Altadena, Calif., U.S.A.

Chemical Analysis of the Surface Layers of Metal Exposed to Different Types of Wear

B. I. Kostetskii, N. L. Golego and P. K. Topekha, (Translated from *Vestnik Mashinostroeniya, 36* (10) (1956) 25–26).*

Specimens with removable surface layers were used in the form of ordinary specimens of 0.80% C tool steel U8A with foil 0.1–0.03 mm thick attached to the rubbing (contact) surfaces. These specimens were tested for frictional wear with the foil surfaces at the friction faces, and then the foil was removed from the surface and used for chemical analysis and other forms of testing.

The composite specimens were abrasion tested under dry friction conditions in air of 60% RH at + 20°C. To assess correctly the effect of adsorptional and diffusional interaction of the metal with the oxygen of the air in abrasion, the content of oxygen in the surface layers of the metal after testing under different conditions was determined.

The white, non-etching layers, characteristic of oxidative wear occupy only 3–4% of the entire cross-sectional area of the removable foil.

The results of gas analyses show a sharp increase in the amount of oxygen in specimens working under oxidative wear conditions (0.5184%). This is almost 30 times that contained in the steel foil prior to wear. Specimens exposed to abrasive wear did not show this increase of oxygen over the initial content; nitrogen of the air does not have any effect.

[1] B. I. Kostetskii, *Wear-resistance of Machine Parts*, Mashgiz, 1950
[2] B. I. Kostetskii, *Fundamental Problems of the Wear Resistance of Machine Parts*, Mashgiz, 1955.
[3] B. I. Kostetskii, V. D. Eliseev and G. N. Preis, *Methods and Machines for the Wear Testing of Metals*, Mashgiz, 1955.
[4] P. K. Topekha, *Main Forms of Metal Wear*, Mashgiz, 1952.

New Method of Increasing the Wear Resistance of Tracks with Open Bushings

A. A. Maurakh, (Translated from *Vestnik Mashinostroeniya, 36* (12) (1956) 16–18).**

The method of increasing the life of links proposed here consists in using bushings of wear-resisting steel pressed into pinbores of LG13 links, the aperture in the pinbores having been previously expanded. In experiments standard track links were used. The pinbores were expanded by forcing a steel ball through them. The bushings were heat treated to high hardness or electrolytically boronized.

We report here the results of using bushings of steels Kh12F1, KhV5, U10 and 45. Those of the first three steels were heat treated, those of St 45 boronized. The surface

* Condensed from Henry Brutcher Technical Translation No. 4037; 2 tables and 2 figures. Price $ 1.90.

** Condensed from Henry Brutcher Technical Translation No. 3962; 4 tables. Price $ 3.85.

hardness of the bushings and pins was 1900–2100 Vickers units, and the thickness of the boronized layer 0.3–0.4 mm.

The hardened bushings of the three alloy steels were run with pins of St 45 with a surface hardness of Rockwell C 58, and the boronized bushings with boronized pins of the same surface hardness as the bushings.

In testing the links with alloy steel bushings a tractor was used for ploughing for about 416 hours, and in testing the links with boronized pins, etc., about 260.

The most wear-resistant of the tested quench-hardened bushings were those made of steel Kh12F1 of the carbide class, although their hardness was lower than those of the bushings from the other alloy steels. The 22.0 mm diameter pins mating with standard pinbores (LG 13, hardness about 170 Brinell units) wore more than the 18.5 mm diameter pins, mating with bushings of high hardness, although the specific pressure in operation was less in the first case. The pins in both cases were of almost identical hardnesses.

TABLE A

Pinbores and corresponding parts of pins	*Average wear in mm of*			
	Boronized pinbores	*Boronized pins*	*Standard pinbores*	*Standard pins*
Intermediate	0.07	0.08	0.88	0.85
End	0.05	0.06	1.15	0.73
Central	0.06	0.05	1.03	0.50

Table A shows that the boronized bushings and pins had extremely high wear resistances. Evidently when the surface layer was considerably harder than the abrasive particles the latter were crushed and did not scratch the boronized surface. However, it was found there is always a small wear of bushings and pins, though no traces of scratching of the boronized surfaces. This may well be due to the formation of oxide films on the surfaces and their removal by the mechanical action of the abrasive during work.

Hence of the steels tested Kh12F1 had the best wear resistance. The wear resistance of bushings of this steel was on the average 6.6 times greater than that of standard links without bushings. The wear of pins mating with bushings of the three alloy steels was in all cases lower than that of the standard pins working in standard links.

Wear Resistance of Steel Surface-impregnated with Carbide-forming Elements

G. N. DUBININ (Translated from *Metallovedenie i Obrabotka Metallov*, (No. 9) (Sept. 1957) 21–25).*

The author studied the wear resistance of carbon steels chromized and treated with other carbide-forming elements, vanadium, columbium, tungsten, molybdenum and manganese. Specimens were gas chromized, etc., by the method already described[3]. Chromizing, vanadizing, and columbizing were effected at 1100°C for six hours,

* Condensed from Henry Brutcher Technical Translation No. 4015; 1 table and 3 figures Price $ 3.50.

tungstenizing and molybdenizing at 1200°C for three hours, manganizing at 800°C for six hours. These conditions were selected to produce carbides of these elements. However, because of high decarburizarion speeds, homogeneous carbides were not formed on the surfaces of the tungstenized and molybdenized specimens, nor on the manganized ones except those containing 1.03% carbon.

Data on specimens chromized under two sets of conditions are given: No. 1: 1100°C for six hours, and No. 2: 1200°C for six hours. Condition No. 2 was used to produce a structure on the surface of the chromized specimens similar to that obtained on the tungstenized and molybdenized specimens (solid solution with precipitation of carbides or intermetallic compounds in grain boundaries). This made it possible to compare the wear resistances of specimens chromized, tungstenized and molybdenized.

The conditions adopted produced in all cases, except steels with 0.15–0.25% carbon after chromizing by the No. 1 condition, a case of sufficient thickness to ensure preservation of the same structure throughout the whole period of wear testing. Wear resistance was determined on a Skoda-Savin machine in 0.5% aqueous potassium chromate made up with distilled water, at 18°C and a 5 kg load. The test lasted for 3000 revolutions of the disc. The control was of steel 18KhGT (nominal 0.18% carbon, 0.50–0.80% manganese, 0.17–0.37% silicon, 0.70–1.00% chromium, max. 0.40% nickel, max. 0.04% S and P each, also Ti), carburized and heat treated in the usual way to a hardness of 780 Vickers units. The relative wear resistance index was the ratio of the wear cavity in the control to that in the investigated specimen. The wear resistances of steels surface-impregnated with columbium and vanadium are about the same, and more than ten times that of the chromized steel. The wear resistance of a chromized steel is about 1.5 times that of a non-chromized steel, confirming previous results[1]. The variation in wear resistance with the carbon content fits into the general pattern.

It was found that in spite of the high surface hardness of steel of different carbon contents chromized by conditions No. 2, tungstenized and molybdenized, the wear resistance of the untreated carbon steel was by comparison considerably higher. The lack of a correlation between surface hardness and wear of the treated steels in these cases is presumably due to the presence of secondary phases at the grain boundaries in the solid solution.

The wear resistance of manganized steel is considerably higher than that of chromized (No. 2 conditions), tungstenized and molybdenized steels. The variation of wear resistance with carbon content generally correlates with changes in hardness, detectable in the individual constituents of the cases. Technical iron (0.03% carbon) having a martensite-like structure (α solid solution) in its surface, has a high wear resistance.

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Friction as Resistance to Shear of Thin Surface Layers on Solids

G. I. EPIFANOV (Institute for Physical Chemistry, U.S.S.R. Academy of Sciences)
(Translated from *Doklady Akad. Nauk S.S.S.R., 114* (4) (1957) 764–767).*

It has been repeatedly suggested that friction is the resistance of the surface layers of solids to shear[4–6]. Up to the present, this viewpoint has not had sound enough experimental foundation. We have worked out a method for studying sliding friction between clean metal surfaces. From a flat specimen of the metal being studied chips of gradually decreasing thicknesses (down to 50–25μ) are cut. Behind the tool a hemispherical block of hardened high speed steel is placed in a special guide, and can be held against the freshly cut surface of the specimen by a normal force. So long as the cut surface does not reach the block, only the cutting force is measured. When the block is lowered gently onto the freshly cut surface, the total cutting and friction forces are measured. Of course, when the specimen emerges from under the tool, the pure force of friction between block and specimen is measured.

In the first series of experiments the relation between the force of friction and the force of adhesion of the block to the surface was studied. The adhesion was determined thus: an oil film was introduced below the unloaded block on the surface and the force needed to move the block in a direction opposite to its original motion was measured. In slow movement of the block the tangential force increased to a certain maximum, and then fell sharply to values corresponding to the friction force in the lubricant.

The results show that the adhesion force is nor more than 15% of the force of dry friction, which indicates that the resistance due to adhesional bridges is only a fraction of the friction force, contrary to what is sometimes maintained[4,5]. The area on which shear actually occurs in friction is considerably greater than the total cross-sectional area of the regions of molecular "seizure". When the block moves on the surface of the specimen, shear occurs not on the contact surface between block and specimen, but inside the specimen at a certain surface.

When the shear area is correctly determined, the frictional pressure is almost identical with the shear strength of the surface layer of metal. If the frictional force at seizure is divided by the area of the friction track, the usual value of the frictional pressure, equal to the shear strength of the metal, is obtained.

Confirmation of these ideas was obtained in experiments with thin metal films, in which the friction of a hemispherical block of hardened high speed steel on steel 1045 was studied, thin layers of copper, zinc or cadmium having first been deposited on the metal.

In complete accord with the findings of other investigators[4,7] thin films of soft metals, deposited on a harder substrate, considerably lower friction force and the coefficient of friction. However, the specific friction pressure in this case is approximately equal to the shear strength of the soft metal of the film, although the friction track lies as before mainly in the hard metal of the substrate. This points to the following conclusions about the mechanism of lubrication.

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* Condensed from Henry Brutcher Technical Translation No. 4327; 2 tables and 1 figure. Price $ 4.90.

When there is a thin lubricating layer of soft metal on a hard surface, or a solid organic film, or, finally, a lubricating layer of metal plasticized by adsorption[8], plastic shear in friction is transferred from the harder metal of the substrate to the soft surface layer, so that the friction track in the harder substrate is formed by plastic compression, as in pure ploughing. The friction force is greatly reduced, becoming equal to the sum of the resistances of the soft surface layer to shear and of the hard substrate to ploughing. When the ploughing force is small, the specific friction force is equal to the shear strength of the surface layer.

The reduction of friction force by lubricant we thus explain not by a decrease of the metal seizure in friction[7,9] but by a change in the nature of the actual process of plastic deformation, leading to a forcing back of metal.

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N. A. Pleteneva and P. A. Rebinder, *Doklady Acad. Nauk S.S.S R.*, *62* (4) (1948) 501-4. (See also Brutcher Translation No. 2365.)
P. A. Rebinder, *Influence of Surface-Active Lubricant-Coolant Liquids on Surface Quality in the Machining and Working of Metals*, U.S.S.R. Academy of Sciences, 1946.
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Received December 11, 1956

Systematic Abstracts of Current Literature

Prepared by Battelle Memorial Institute, Columbus, Ohio, U. S. A.

Reprinted from Battelle Technical Review 1958

1. Deformation and Fracture

Aspect rhéologique de la dureté.
Rheological Aspect of Hardness.
A. Braun. *Revue de Métallurgie*, v. 55, no. 5, May 1958, p. 470-474.
Three categories of the properties of solids are distinguishable: elasticity, plasticity, and viscosity. Analysis of the behavior of local deformations set up by stresses to which the solid offers a resistance, which is designated hardness, shows the necessity of defining hardnesses related to the three categories mentioned above. Their significance flows from their relation with the respective constants of elastic, plastic, or viscous matter, a relation which the author calls "hardness function" and which shows also whether these hardnesses are dependent on experimental conditions. This conception of specific hardness allows one to envisage a more detailed analysis of the various rheological properties of a solid, by means of appropriate hardness measurements and by making use of rheological models.

Theoretical Principles in Metal Shaping.
(in Czech)
Antonín Vach. *Hutník (Czech)* v. 8, no. 6, June 1958, p. 203-206.
Deformation of monocrystals. Study of plastic, sliding, and diffusion deformations. Basic laws of plastic deformations. Problems of strengthening and recrystallization, resistance to deformation, distribution and magnitude of main stresses, and forging.

Dislocation Theory of Fracture of Crystals.
F. E. Fujita. *Acta Metallurgica*, v. 6, Aug. 1958, p. 543-551.
Formation of an incipient small crack or flat hollow dislocation; theory of elasticity of slender hollow dislocation and application of theory to the fracture model; energy of the system as a criterion of fracture condition.

Transgranular and Intergranular Fracture of Ingot Iron During Creep.
L. A. Shepard and W. H. Giedt. *U. S. National Advisory Committee for Aeronautics, Technical Note* 4285, Aug. 1958, 26 pp.
Concludes that a vacancy-condensation mechanism is not probably responsible for high-temperature intergranular fracturing in ingot iron.

Fracture Mechanism of Semi-Brittle Steels in Torsion.
J. B. Hunt. *Journal of the Mechanics and Physics of Solids*, v. 6, no. 4, July 1958, p. 321-326 + 1 plate.
The petal-shaped pattern noticed on the fracture surface of certain mild steels in a type of fracture hitherto assumed to be a conventional shear separation. It was found that the fracture sequence involves both tensile failure and sliding in shear.

The Propagation of Cracks and the Energy of Elastic Deformation.
H. F. Bueckner. *ASME, Transactions*, v. 80, no. 6, Aug. 1958, p. 1225-1230.
Reviews crack extension energy and strain energy on a 3-dimensional basis, and shows explicitly the applicability of classic theorems by Betti and Clapeyron. One of the results is that the energy supplied for crack extension equals the strain energy of the difference of the two stress fields before and after crack extension.

Correlations of Brittle-Fracture Service Failures With Laboratory Notch-Ductility Tests.
P. P. Puzak, A. J. Babecki, and W. S. Pellini. *Welding Journal*, v. 37, Sept. 1958, p. 391s-410s.
Nil-ductility transitions determined by drop-weight tests are demonstrated to correlate with the service performance of the steels investigated.

Sudden Fracture of Machine Parts and Structure Elements.
G. V. Uzhik, M. J. Galperin, and A. A. Zooykova. Paper from "Symposium on Large Fatigue Testing Machines and Their Results" p. 132-141, 1958. American Society for Testing Materials, Philadelphia.
Transition of fatigue failure to sudden brittle fracture.

2. Friction (see Authors' Abstracts)

3. Lubrication and Lubricants

The Effect of Turbulence on Motion in the Lubricant Layer. (in Rumanian)
V. N. Constantinescu. *Studii și Cercetări de Mecanică Aplicată*. v. 9, no. 1, 1958, p. 103-137.
Equations of lubrication under turbulent conditions, starting from the general Reynolds equations. Distribution of velocities based on the hypothesis of Prandtl concerning the length of turbulent mixture.

Lubrication of Titanium.
Nicholas Fatica. *U. S. Air Force, Wright Air Development Center, WADC Technical Report* 57-61, pt. 2, May 1958, 78 pp.
Information about the frictional properties of modified Ti coatings, and a comparison of the wear resistance of the best surface treatments in the presence of various lubricants. The use of conventional lubricants for any but electroless nickel and carburized iron-plated Ti is unreliable. Halogenated materials appear to be the best lubricants for the other treatments.

Effect of Metals on Lubricants.
John B. Christian. *U. S. Air Force, Wright Air Development Center, WADC Technical Report* 54-576, pt. 3, May 1958, 23 pp.
Shows the effects which an ester and a disiloxane have on various metals, and the effects which those metals simultaneously have on the fluids at elevated temperature (550° F). The disiloxane proved to be compatible with all of the metals tested, while the ester showed signs of being generally incompatible with all of the metals.

On the Boundary Lubricity of Non-Polar Hydrocarbon Oils and Their Mixtures.
Yasukatsu Tamai. *Institute of Petroleum, Journal*, v. 44, July 1958, p. 207-211.
Lubricating properties of benzene, cetane, decalin, and di-*cyclo*-hexyl methane.

4. Bearings

Bearings, Lubricants and Lubrication.
Mechanical Engineering, v. 80, Sept. 1958, p. 64-74.
A digest of 1957 literature covering over 200 references. Includes metalworking lubricants.

Design Data for Flat Air Bearings Under Steady Load Conditions.
H. L. Wunsch. *Engineer*, v. 206, Sept. 12, 1958, p. 411-415.
While it is not too difficult to design an air bearing for a particular purpose and to make it work, it is by no means so easy to get the optimum design from the point of view of maximum stiffness and load, and economic air flow.

Flexible Bearings.
Automobile Engineer, v. 48, no. 7, July 1958, p. 255-259.
Fundamental principles involved in the employment of rubber and nylon bushings.

Plastic Ball and Roller Bearings.
Joseph E. Montalbano. *Machine Design*, v. 30, Aug. 7, 1958, p. 96-99.
Advantages, applications, and limitations.

Pressure-Fed Bearings: A Designer's Guide.
Donald J. Myatt. *Machine Design*, v. 30, Sept. 18, 1958, p. 161-164.
A practical engineering approach to balancing the many variables involved.

5. Wear and Wear Resistance (see Authors' Abstracts)

6. Analysis and Testing

Scratch Hardness. II. X-Ray Investigation of Structural Change.
Tomiya Sutoki and Koichi Nakajima. *Tôhoku University, Science Reports of the Research Institutes, Ser. A. Physics, Chemistry, Metallurgy*, v. 10, no. 4, Aug. 1958, p. 269-275.
Structural change due to scratching or indenting Al and Ag was studied by X-ray analysis. It may be reasonably interpreted that scratch hardness of a metal is related to the annealed state. This is contrary to Tammann's view that scratch hardness is related to the state of maximum hardening, being independent of previous history of the metal.

An Index of Cavitation Erosion by Means of Radio-isotopes.
S. L. Kerr and Kjell Rosenberg. *ASME, Transactions*, v. 80, no. 6, Aug. 1958, p. 1308-1314.
Use of an insoluble radioactive coating on a runner blade in a hydraulic turbine.

Measurement of Casting Surface-Roughness.
C. H. Good and C. E. McQuiston. *Foundry Trade Journal*, v. 105, Aug. 14, 1958, p. 191-195.
The various terms used in the study of surface roughness are defined and a survey is given of the methods available for evaluation and control of this factor.

7. Surface Treatment and Finishing

New Hard Coat for Aluminum.
Steel, v. 143, Sept. 22, 1958, p. 80-81.
The electrochemical method can deposit coatings more than 0.006-in. thick, has better wear and heat resistance.

Surface Finish. Metallurgical and Mechanical Aspects.
K. G. Lewis. *Metal Treatment and Drop Forging*, v. 25, Aug. 1958, p. 335-342.
The various expressions used in the study of surface finish are described, together with the more important machine and material variables which affect them. Particular attention is paid to chip formation and its influence on surface finish. The effect of metallographic structure is examined at length owing to its bearing on the formation of built-up-edge. Investigations into the mechanism underlying the formation of "smear metal" are described and the various means for its removal compared, the comparison including an examination of the effect of cold-working on surface finish. (To be continued.)

Fretting Corrosion of Large Shafts as Influenced by Surface Treatments.
Oscar J. Horger and H. R. Neifert. Paper from "Symposium on Large Fatigue Testing Machines and Their Results". p. 81-95, 1958. American Society for Testing Materials, Philadelphia.
Rotating-bending fatigue tests were made on 7-9/16 in. diameter normalized-and-tempered 0.50% plain carbon steel shaft forgings having a pressfitted outer disk member. Information is presented on the actual surface rolling of about 27-in. diameter marine propeller shafting.

Effect of Some Physico-Mechanical Properties of Brittle Materials on the Process of Their Polishing. (in Russian)
F. K. Aleinikov. *Zhurnal Teknicheskoĭ Fiziki*, v. 27, no. 12, Dec. 1957, p. 2725-2737.
Effect of predominant properties of marble, various glasses, natural crystal, ruby, and other materials on polishing characteristics such as productivity, width of relief, and crumbling layers.

New Abrasives: New Grinding Techniques.
K. McIndoe. *Australasian Engineer*, 1958, June 7, p. 107 + 8 pages.
Cutting theory; grinding wheels; types of surface finish; sharpening and dressing equipment and tools.

Shot-Blasting in the Shipyard.
Corrosion Technology, v. 5, Sept. 1958, p. 285-287.
Describes the development and application of a new shot-blasting installation which has been specially designed for use on ships' plates.

Two Shipyards Install First U. K. Airless Shot Blast Plants.
Industrial Finishing (London), v. 10, Aug. 1958, p. 27-29.
Two airless shot blast installations in shipyards are believed to be the first machines of this kind to be put into operation in the U.K.

A Treatise on the Internal Mechanics of Ball, Tube and Rod Mills.
H. E. Rose and R. M. E. Sullivan. 258 pp. 1958. Constable & Co., London. (TJ1345 R72t)
Analyzes motion of mill charge, power requirements, grinding processes, wear of grinding media and mill linings, and role of additives in the mill charge.

8. Machining and Tool Wear

The Fundamentals of Wire Drawing.
J. G. Wistreich. *Metallurgical Reviews*, v. 3, no. 10, 1958, p. 97-142 + 2 plates.
Mechanics; effects of heat and temperature; wire finish, friction, and die wear; lubrication; wire properties.

Modern Cutting Tools Save Costs At High and Low Speeds.
H. J. Siekmann. *Iron Age*, v. 182, no. 12, Sept. 18, 1958, p. 95-97.
The use of new cutting tool materials at high machining speeds results in substantial savings in operating costs. In many cases, however, low speeds are more efficient and economical.

Ceramics—A Material for Gages?
Carl G. Erickson. *Carbide Engineering*, v. 10, Aug. 1958, p. 13-15.
Ceramics are being applied to regular conventional cylindrical plug and ring gages, thread plug gages, and various wear applications such as anvils for electrolimit comparators.

Bibliographies

Dictionaries

International Bibliography of Dictionaries and Glossaries of Petroleum, Lubrication and Related Subjects, Typescript 5 pp. The Overseas Translation Service, Inc., Dictionary Procurement Center, P. O. Box 402, Stamford, Conn., U.S.A.

Technical Translations

List of Available Catalog Sections, issued 1956, 8 pp., several hundred sections. Henry Brutcher Technical Translations, P. O. Box 157.

The following lists are of special interest to readers of *WEAR*.

List C-236: Wear behavior of ferrous materials, 10 pp.; Translations of (essentially) German and Russian papers published between 1934 and 1955. Title, short abstract and price of translation are given.

List C-284: *Testing for wear resistance*, 3 pp.; 8 Russian and 4 German papers.

(See also Abstracts Section of this issue: Recent Russian Literature.)

From NACA to NASA

Research Abstracts and Reclassification Notices, formerly issued by NACA, will no longer be issued. Publications Announcements will be issued by NASA at intervals of 2–4 weeks. NACA documents will be controlled by NASA, which will furnish NACA documents that are still available. The practice of keeping documents "in print" for 5 years will be followed, for both NASA and NACA documents.

Research information released by NASA will be distributed in three series of documents:

Reports	(unclassified only)
Memorandums	(classified and unclassified)
Republications	(classified and unclassified)

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Government Printing Office
Washington 25, D. C.

Memorandums and Republications will be available from:
Division of Research Information
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
1520 H Street, N. W., Washington 25, D. C.

Joint SAE and API Research Projects

Annual Report 1958, Coordinating Research Council, Inc., 30 Rockefeller Plaza, New York 20, N.Y., 146 pp. Sustaining members: SAE and API.

Detailed reports of CRC projects on:

Fuel, Lubricant (and Wear) and Equipment Research

in the fields of aviation, diesel vehicles and motor vehicles. Lists of new projects, completed projects, released reports and membership.

Recent and Forthcoming Events

American Society of Lubrication Engineers
84 E. Randolph Street, Chicago, Ill. U.S.A.

Gear Symposium

January 25th–27th 1959, Chicago, Hotel Morrison

ASLE Annual Meetings and Exhibits

April 21st–23rd, 1959, Buffalo, New York, Hotel Statler

ASLE/ASME Joint Lubrication Conference

October 19th–21st, 1959, 6th Conference, New York, Sheraton-McAlpin Hotel

C.I.M.A.C. Congress (Congres International des Machines à Combustion)

June 14th–19th, 1959, Wiesbaden, Germany

Subject: *"Diesel Engines and Gas Turbines up to 1,500 H.P.– Current problems concerning design, production, development and running at site"*
Further information obtainable from National Committees.
Address of British National Committee: 6 Grafton Street, London, W.1.
Address of Deutsches Nationales Komitee: Frankfurt/Main, Brentanostr. 29.

Third European Congress of Petroleum Equipment

June 22nd–25th, 1959, Paris

This congress is being organised by the Federation of European Petroleum Equipment Manufacturers and will be under the Direction of Mr. R. NAVARRE, Chairman of the Institut Français du Pétrole.

23 experts from different countries, representing suppliers or users of petroleum equipment, will present technical papers on *inter alia:* Drilling techniques and equipment; the outcome of the Fifth World Petroleum Congress held in New York.

Secrétariat Général, 44 bis, Rue Pasquier, Paris-VIIIe.

Notes on Contributors

D. F. DENNY: born near London; awarded London degrees B.Sc. (Eng.) in 1944, Ph.D. in 1949. Has been on the staff of the British Hydromechanics Research Association since 1949, where his work has included study of fluid seals and bearings; author of a number of papers on friction of rubber and on fluid seals. [See p. 264]

H. L. HUGHES: born in Birmingham; educated at Kings Norton Secondary School, and subsequently at the Central Technical College as a part-time student; graduated in the Faculty of Engineering of London University in 1935 as an external student. In 1930 joined the staff of the then City of Birmingham Electric Supply Department and held a series of engineering appointments of gradually increasing responsibility; in 1947 appointed to the teaching staff of the College of Technology, Wolverhampton, and now holds the position of senior lecturer in the Department of Electrical Engineering. [See p. 311]

H. D. LEES: obtained his degree in mechanical engineering at the Technical University of Copenhagen in 1941; in that year joined Helsingør Skibsvaerft og Maskinbyggeri A/S (The Shipyard and Engine-Builders of Elsinore); in 1945 joined D.F.D.S. (The United Steamship Company), Copenhagen, being appointed chief engineer in 1948 and vice president in 1956. [See p. 273]

R. M. MATVEEVSKY: born in Viatka (U.S.S.R.); graduated in mechanical engineering at the Textile Institute, Moscow in 1914; in 1953 received degree of C.Sc. (in technology) from the Institute for the Study of Machines, Academy of Sciences of the U.S.S.R.; now a senior scientific officer at the Laboratory of Wear Resistance of the same Institute, being especially engaged in studying friction and wear under boundary lubrication conditions; author of the monograph *A Temperature Method for the Estimation of Limiting Machine Oils Lubricity* (Academy Press, U.S.S.R., 1956) and of several papers on that subject. [See p. 315]

C. RUBENSTEIN: (for biographical note see *Wear*, *2* (1958/59) 164). [See p. 296]

FRANK W. SMITH, B.A., PH.D.: born in Wales; studied chemistry at Oxford and Dundee; presently employed in the Mechanical Engineering Division of the National Research Council of Canada at Ottawa. [See p. 250]

V. ŠOLAJA: (for biographical note see *Wear*, *2* (1958/59) 84). [See p. 311]

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(b) Surveys of work that has already been published either in languages other than those of this Journal or in reports on laboratory activities that are not generally accessible.
(c) Reviews on recent developments in specialized fields.
(d) Case histories or analyses of service problems, if they are of a more general interest.
(e) Short notes or letters to the editors.
(f) Authors' abstracts, Bibliographies and Systematic abstracts.

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